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LOWER COST PACKAGING FOR GUIDED MISSILE ELECTRONICS

Kenneth E. Woodward

28 March 1961

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ABSTRACT

A study was made of packaging techniques for guided missile electronics in which it was pointed out that the real cost of such electronic devices is not that cost required to manufacture the end product, but that cost which includes its share of the financial burden of development and production. It was shown that the overall costs could be reduced by shortening development programs through the elimination of steps in the packaging process and by substantially reducing the effects of the environment so that less costly electronic components can be utilized. The method is dependent upon the use of (1) mounting materials with certain specific qualities to insure optimum performance of packages in several simultaneous environmental situations and, if necessary, (2) cooling fluids compatible with anticipated heat inputs and stated engineering boundaries.

1. THE PROBLEM

The packaging of electronic equipment for guided missile applications is expensive. The chief factors contributing to the high costs are the severe environments in which the devices must operate and the required reliabilities. Customary packaging approaches usually emphasize the achievement of these two objectives with the costs relegated to positions of lesser importance, and rightly so; however, costs cannot be completely ignored.

Minimizing packaging costs when only vibrational stresses are present is fairly simple, but when shock is added the cost problem is complicated. It is further complicated with the introduction of noise, sustained acceleration, and temperature stresses. Equally essential to successful design, and not necessarily associated with environmental stresses are weight, size, shape, and sealability of the package.

It is obviously not possible to design optimally for environment and high reliability while simultaneously incurring minimum packaging costs. However, it might be possible to design a package that satisfactorily meets environmental and reliability requirements and yet seriously attempts to reduce costs.

Therefore it is the purpose of this study:

(1) To examine briefly present techniques used in packaging guided missile electronics for their adequacy in meeting the demands of current and future missile environments.

(2) To explore the engineering and cost parameters in the development of small guided missile electronic packages with the intention of developing a set of design parameters that would possibly lower packaging costs while simultaneously permitting the packages to adequately withstand present and future missile environments.

(3) To demonstrate the evolved parameters by describing the tests made on a model incorporating these in its design.

To illustrate the difficulty of satisfying environmental objectives and the resulting cost burden, figure 1 shows the various design phases through which one package passed before achieving a design capable of meeting the missile environment. Models A, B, and C failed to meet environmental objectives because of undesirable resonances resulting either from inadequate design or from expanded frequency ranges associated with improved knowledge of missile performance. By incasing the modules of C in a cast aluminum housing, a marginal unit, D, resulted. Model E was the final satisfactory development item. The entire development program was performed primarily to permit the fuze circuitry to withstand, and operate reliably in, the environment imposed by the missile. Five years were required to complete the program (including several test equipments) at a cost of $3,000,000. Additional money and time were required to engineer model E for quantity production.

Redesign for the environment is not an infrequent occurrence. The environment was a major cause for redesign of the initial models of the proximity fuzes for the Redstone, Sparrow I, Terrier, Corporal, and Honest John missiles.

This situation emphasizes the need for less costly packaging techniques that can satisfy environmental requirements.

The scope of the present study has been restricted to small electronic devices because of the greater ease of attaining the proper structural resonances required by subsequent design considerations. The upper bound to small packages cannot be precisely defined other than to restrict it to those meeting the design objectives set forth herein. For example, equipment weighing 10 to 15 lb would have little difficulty in conforming with the proposed objectives as opposed to those weighing 100 lb or more.

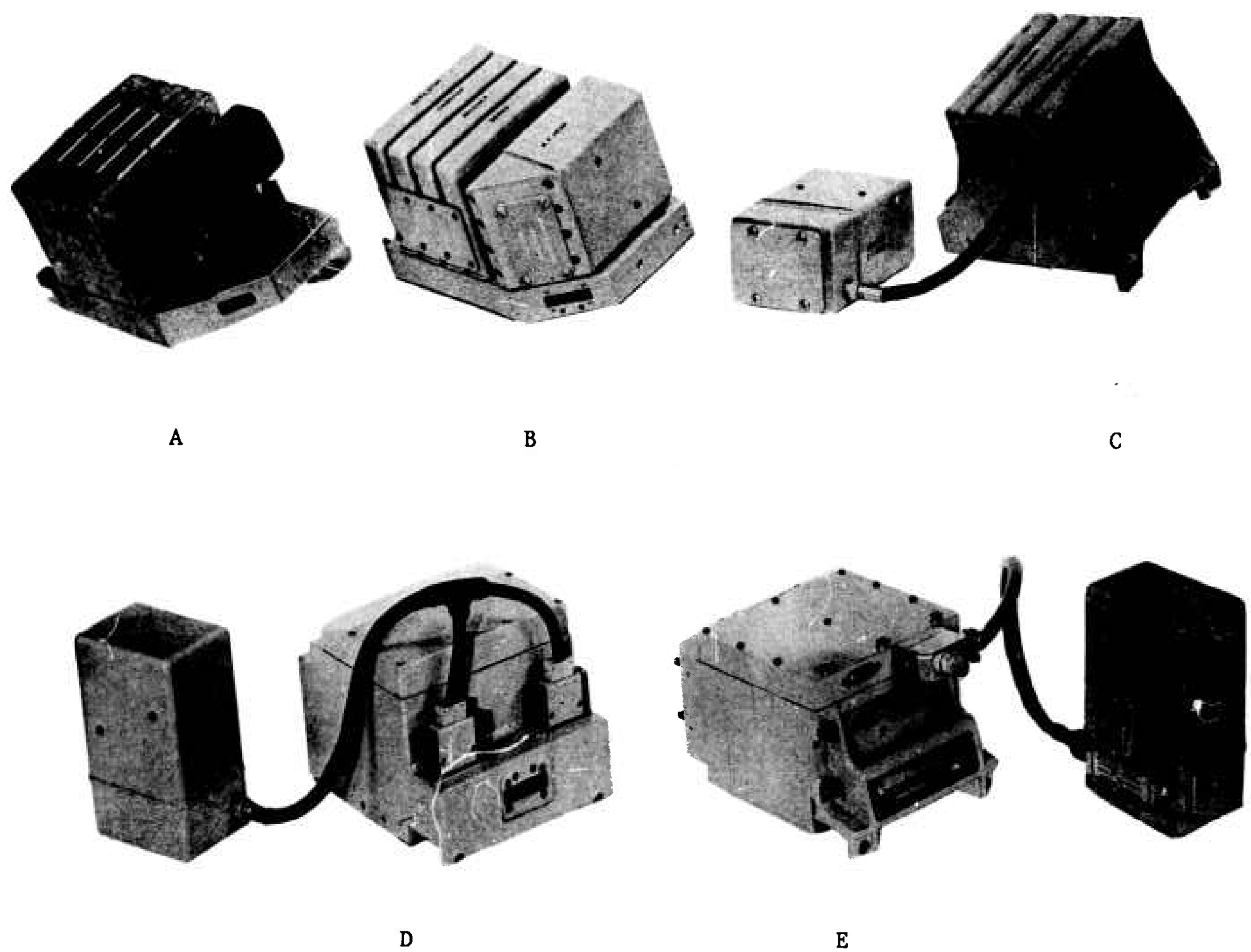

Figure 1. Evolution of a fuze

2. THE ENGINEERING BOUNDARIES

Military equipment must be capable of sustained operations under extreme environmental circumstances. These environments impress a variety of stresses on the equipment induced by either dynamic forces and/or thermal conditions. In addition, deterioration or failure of the equipment may be caused by other destructive influences such as corrosion deposits from salt spray or fungus growth, or, in desert climates, sand and dust, which may work into delicate moving parts. Recently nuclear weapons have added the effects of radiation to the already lengthy list of problems the designer must consider.

It is difficult in the early planning stages of the missile project to know exactly what environment the components must withstand. Without precise knowledge especially with regard to the dynamic situation, it is the usual practice to use military specifications (or other governmental publications) as design criteria. It is important for military specifications to be a part of the formal agreements between the government and contractor or between the contractor and subcontractors to define the legal conditions of acceptance. Military specifications define the conditions under which satisfactory operation must occur and hence establish a standard for the measurement of operational improvement. Further, the tests conform to readily available test apparatus. Obviously, it is assumed that equipment "passing the specs" will result in a satisfactory item, and this is the primary reason for any specification.

2.1 Summary of Environmental Requirements

The most recent and concise summary of the environmental requirements for the three branches of the military services has been prepared by the Office of the Director of Defense Research and Engineering into a guide (ref 1). The guide encompasses not only the usual military specifications but also considers the newer areas of nuclear propulsion. Its purpose is "for use in planning current and future research and development programs involving electronic component parts." The tests by which the requirements are applied are detailed either in the document or in referenced military specifications.

The guide defines electronic component parts to include basic circuit elements such as capacitors, resistors, switches, relays, transformers, crystals, waveguides, electron tubes, and semiconductors devices, and specifically excludes complete equipments or subassemblies. It is an objective of the guide to describe conditions under which not only the prime equipments in which the parts are placed must operate but also those environments in which they are placed within the equipment. Table 1, a reproduction of applicable portions of a table in the guide, summarizes the environments for four groups of electronic component parts used in guided missiles and high-performance aircraft. The groups of equipment are defined as follows.

"Group V covers that group of electronic component parts for use in electronic equipment of high-performance aircraft and specialized shipboard applications.

"Group VI covers that group of electronic component parts for use in electronic equipment of nuclear-powered aircraft and ballistic missiles.

"Group VII covers that group of electronic component parts in specialized applications in electronic equipment of high-performance aircraft and missiles.

"Group VIII covers that group of electronic component parts for use in electronic equipment of nuclear powered weapons."

The requirements set forth in table 1 and the tests described in the guide and in military specifications generally are not exact representations of actual service conditions. Rather, they represent a compromise between actual service conditions in the field and the simulated field-service conditions obtainable in the laboratory.

2.2 Package Characteristics

In the design of the package and its contents to withstand the environmental domain the combined effect of the environmental stresses on the package and each of its components must be considered. But there are other design considerations equally important to the success of the design and not necessarily associated with stress. For an adequate design the package must be:

1. Easily producible. The use of simple standard structural shapes, off-the-shelf components, etc, reduces costs, expedites development and production schedules, and simplifies maintenance.

2. Adaptable to a variety of configurations. The cramped spaces in current missiles do not always lend themselves to simple solid shapes.

Table 1. Environmental Requirements Chart

Environmental characteristics	Group V	Group VI	Group VII	Group VIII
Temperature:				
Operating	.-65° + 125°C	-65° + 200°C	-65° + 200°C	-65° + 350°C
Storage	-65° + 71°C	-65° + 71°C	-65° + 71°C	-65° + 71°C
Pressure:				
Operating	1.32" Hg	0.315" Hg	0.04" Hg	0.315" Hg
Altitude (ft)	70,000	100,000	150,000	100,000
Nonoperating	NA*	NA	NA	NA
Altitude (ft)				
Moisture (100 % relative humidity)	10 c	10 c	10 c	10 c
Vibration:				
Cycles per second	10 - 2,000	10 - 2,000	10 - 2,000	10 - 2,000
Acceleration (g)	10	15	15	20
Shock:				
Acceleration (g)	50	50	50	50
Time in milliseconds	11 ± 1	11 ± 1	11 ± 1	30 ± 1
Air-induced vibration:				
Cycles per second	150 - 9,600	150 - 9,600	150 - 9,600	150 - 9,600
Db above 2×10^{-4} dynes/sq cm	160	160	160	160
Acceleration:				
(Constant) g	50	50	50	50
Time in seconds	10	10	10	10
Explosive atmosphere	**	**	**	NA
Nuclear radiation:				
Neutron flux level (fast)				
Neutron/cm^2-sec		NA	10^9	NA
Time in hours		NA	1,000	NA
Gamma photon flux level				
Photon/cm^2-sec		NA	10^{11}	NA
Time in hours		NA	1,000	NA
Thermal neutrons			***	
Sand and dust	**	NA	**	**
Salt atmosphere per MIL-STD-202 (hr)	96	96	96	96
Flammability				NA
Fungus resistance	Non-nutrient in all groups			
Life (hrs)				
Operating	2K	20K	2K	2K
Storage	30K	30K	30K	30K

NOTES: *Not applicable.

** The part is to be tested in accordance with Procedure I of specification MIL-E-5272.

*** Thermal neutrons are not listed as a requirement, but, since all neutron fluxes have some thermal component, this component should be measured and reported with all tests.

3. Lightweight with good volumetric efficiency (high component density). Unnecessarily massive structures reduce missile payload. A package should be made as small and light as possible. Package densities of 0.03 lb/in.^{3} or less* are obtainable in missile packaging.

4. Sealable and capable of being pressurized. The rapid altitude transition and the extended storage possibilities under degrading atmospheres require this of the package.

5. Longlived under adverse handling and storage conditions. Five-year storage (ref 1) capability is commonly required of electronic missile hardware.

6. Capable of flight attitudes at any angle during missile maneuvering.

7. Operable under sustained steady-state flight accelerations occurring at blast-off and during maneuvering.

8. Resistant to corrosive atmospheres created by salt spray and rocket propellants.

2.3 The Character of Electronic Components

Several rather general observations should be made about the vibrational responses of electronic components such as vacuum tubes, relays, etc, and these components are equally a part of the engineering boundaries for it is these with which the designer must live in the environments previously described. Normally, it is relatively easy to obtain and mount commercial components having structural resonances above 400 or 500 cps. But for guided missile frequencies higher than these, adequate commercial components are difficult to obtain. The general classes of subminiature tubes as used in guided missile fuzing, for example, develop fundamental filament resonances starting at about 800 cps and continuing through 1400 cps (ref 2), which is within the range of present vibration specifications.

The Armour Research Foundation of the Illinois Institute of Technology (ref 3), in a contract with the Air Materiel Command of the Air Force to eatablish a standard type of vibration measurement test procedure for electron tubes used in high-performance aircraft and guided missiles, has found for the limited number of tubes investigated that major resonances

* An objective for the fuze shown in figure 1.

of the miniature tubes occur above 500 cps. Major resonances of the subminiature types are above 1000 cps. Transmissibilities of the resonances (ratio of system response to exciting vibrations) vary between 12 and 800:1. These results were for inputs not exceeding 2 g. Similar resonant characteristics are reported for the specific subminiature tubes used in guided missile fuzes. The general types of subminiature tubes used in the DOFL fuzes develop both filament and cathode resonances of importance above 500 cps (ref 4).

Terminal boards and other mountings for resistors and condensers can be easily stiffened to produce resonances somewhat above 400 or 500 cps. Higher resonant frequencies require proportionately more effort. But the important consideration is that the region below 500 cps can be generally cleared of damaging mounting resonances, for small items.

It is unfortunate that the resonant characteristics of such components as tubes and relays are not completely under the control of the packaging engineer. While their mountings can be adjusted in stiffness, the sealed elements of the components cannot be usually controlled. From table 1, the environmental frequencies extend to 2000 cps or higher, well above the frequency at which uncontrollable component resonances frequently start.

This situation suggests that isolation (vibrational attenuation) of the higher exciting frequencies is desirable especially in view of the packaging engineer's lack of control over certain resonant characteristics of fuze components, and because of the multitude of resonances occurring at these higher frequencies. Isolation would restore a large measure of control to the designer with attendant improvements in reliability and performance predictability.

3. TRENDS IN PRESENT PACKAGING

The methods used to mount and package guided missile electronic equipment are usually influenced most by their dynamic environments. The techniques for controlling the nondynamic environments, such as temperature, usually do not conflict with the approaches for controlling vibration and shock. To survive the dynamic environments, two methods are most commonly used, popularly termed "isolation" and "stiff" packaging.

3.1 Isolation Packaging

C. E. Crede (ref 5, p 146) describes an isolator "as a load-supporting resilient element having controlled elasticity and damping, and adapted to be integrated into a mechanical or structural assembly." In electronic equipment mounted on missile structures violently vibrating at a multitude of frequencies and amplitudes, isolators are used to reduce the motions experienced by the supporting structure.

Most commercially available isolators have in the past caused the mounting resonant frequency of the isolated equipment to occur at less than 15 cps, because of military specification requirements (MIL-C-172B). But recent considerations in isolator design have increased this frequency to 20 to 30 cps for loadings in any direction. A sacrifice in vibration isolation properties was permitted so that some shock isolation could be achieved. For example, Barry Controls, Inc, markets a mount under the trade name All-Angl isolator (ref 6), which produces mounting resonances of about 24 cps with transmissibilities limited to 1.5:1 at resonance. The isolator can handle fairly large input vibrational displacements at resonance without bottoming (i.e., without becoming excessively stiff). This trend toward higher isolation frequencies is engendered by the desire to develop a mount that can handle moderate shock impulses (30 g) encountered in handling and use.

For applications where high impact shock can be expected, such as on shipboard, snubbing-type isolators and compressed rubber mounts are used in which resonance occurs at somewhat higher frequency (ref 7), with isolation taking place above 45 cps or so. These mounts operate on the principle of storing energy imparted under a sudden velocity change and releasing it gradually over a longer period of time, thereby reducing peak accelerations. The snubber can tolerate several hundred g in contrast to 30 g for the lower-frequency isolator, but the transmissibilities at resonance are considerably higher.

Figure 2 depicts several of the more customary methods of attaching isolators to the equipment. For equipment mounted as in figure 2 (a), horizontal forces acting through the center of gravity cause not only a horizontal displacement but a rotational displacement as well. The situation is called "coupling". Mountings (b), (c), and (d) demonstrate isolator arrangements that exhibit decoupled modes of vibration, and those are the most desirable for controlling multi-directional vibrations. The system shown in (d) is, however, critical

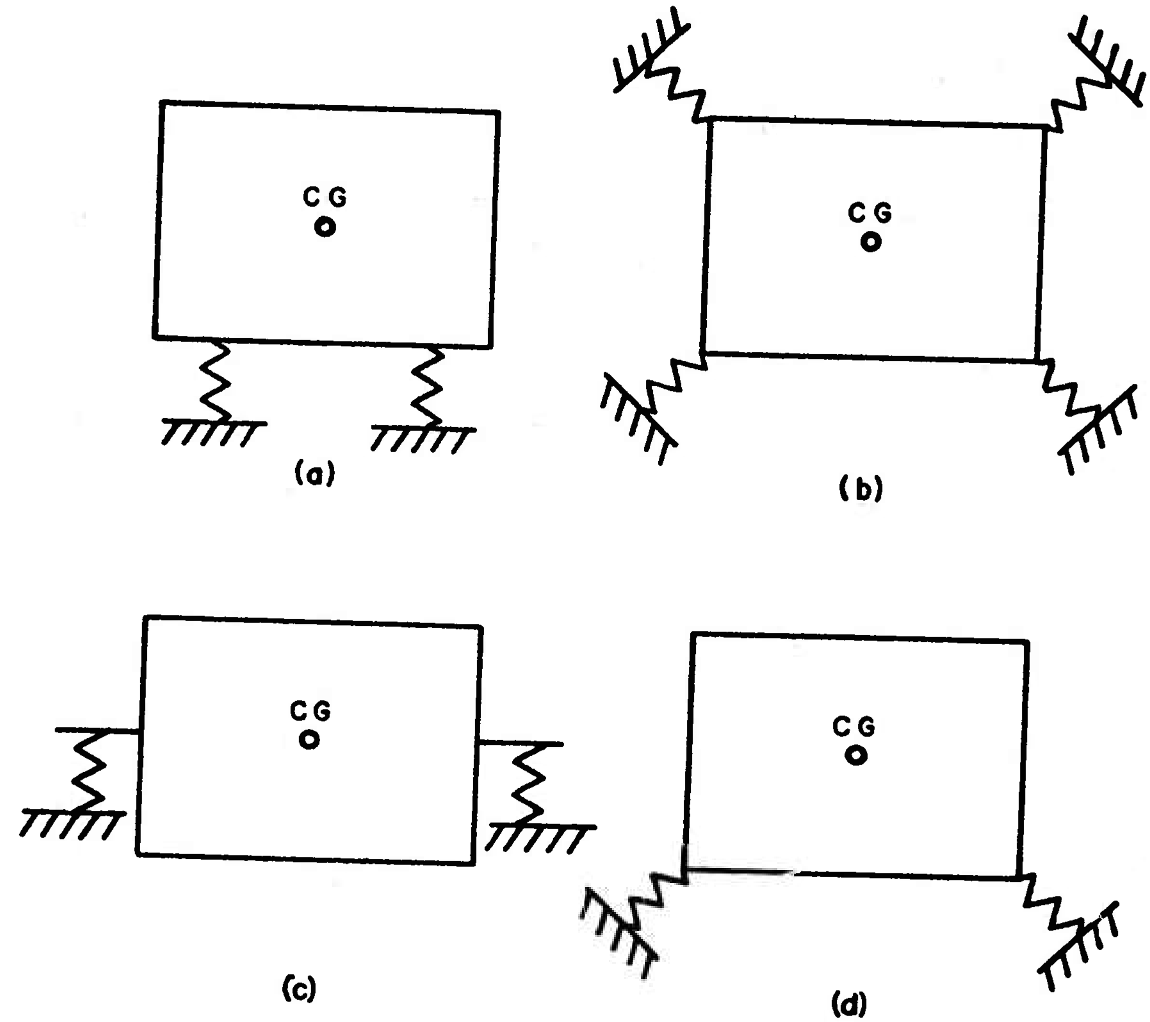

Figure 2. Typical isolator mounting arrangements

with respect to the physical parameters if true decoupling is to be attained.

Generally, isolators are fairly extravagant with space both for that required for the isolators themselves and the additional space necessary to permit bouncing of the isolated equipment. This, together with the fact that they can withstand relatively small sustained accelerations without bottoming, causes many missile designers to use stiff packaging techniques.

3.2 Stiff Packaging

The objective in stiff packaging is to cause all resonances of the equipment to lie above the highest exciting frequency of the environment or test specification. (For real missile environments, this can not generally be accomplished.) This is opposed to isolation techniques in which the basic resonance of the mounting is forced below (or nearly below) the lowest exciting frequency. The purpose of the stiff approach is to prevent excessive amplifications of vibrations of both the package and its components within the test spectrum. The fuze shown in figure 4 is an example of this type of packaging. While for relatively lightweight packages (or fuzes) it is possible to desigh for moderate maximum frequencies of 1500 or 2000 cps, it becomes increasingly more difficult as the weight and size of the equipment increase.

Stiff packaging techniques have the advantage of conserving space (none needed for isolators and bouncing). Under conditions of sustained acceleration such as at blast off and during high-velocity maneuvering, stiff designs have little difficulty in contrast to isolation systems which tend to bottom. This absence of bouncing and bottoming also has beneficial effects in missile guidance by eliminating certain inertia effects. For these reasons, stiff packaging is favorably considered particularly for high-velocity missiles. It also has the advantage of preventing amplification of the shock experienced by the basic structure, but it does not attenuate shock for the customary pulses defined by military specifications (table 1).

Stiff packages are usually heavier than isolated packages. Several of the successfully packaged DOFL fuzes have densities approaching that of solid aluminum (0.1 lb/in.^{3}). Stiff packaging generally requires longer development programs because of difficulty in raising the many resonances above the upper frequency boundary.

3.3 Resonances Within The Frequency Spectrum

The alternative to isolation in which the equipment is caused to resonate near or below the low end of the spectrum and stiff packaging where resonance are forced above the upper spectrum extreme is to let resonances occur within the spectrum. This while normally thought of as undersirable, is often unavoidable, especially in large complex structures.

If resonances do occur in the spectrum, the designer usually tries to avoid predominant missile resonances, if such are known, by causing the equipment to resonate elsewhere. Similarly, equipment structural resonances are forced away from component resonances. Of particular importance is damping unavoidable resonances to amplitudes that the equipment can tolerate.

4. DESIGN CONSIDERATIONS FOR ECONOMY

Opportunities for economy are presented in both the development and production phases of missile electronic hardware. Development costs usually represent a significant portion of the total cost of a procurement program where the quantity of production is severely limited by the rapid rates of missile obsolescence. Packaging the hardware so that it not only satisfies the environmental requirements but also shortens the development program saves recognizable sums of money. Production economies can be accomplished by a reduction of the environmental stresses experienced by the components.

First consider the area of development.

4.1 Reduction of Development Costs by Performance Predictability

From a cost viewpoint the ability to predict system or package performance prior to the actual construction of hardware is important. Systems that perform in accordance with precise, simply defined laws are desirable because the system's ability to remain within the range of operating characteristics or design objectives can be quickly established. As a byproduct, costs are reduced because the testing necessary to determine the various parameters of performance is significantly reduced.

However, predicting performance prior to system fabrication is not simple. To predict, for example, the complete vibrational response of a system having six degrees of freedom, i. e., three translational and three rotational, requires the simultaneous solution of six equations of the form:

$$m\ddot{x} - cf(\dot{x}) - kf(x) = f(t)$$

where $m\ddot{x}$ defines the inertia force, $cf(\dot{x})$ the damping force, $kf(x)$ the restoring force, and $f(t)$ the disturbing force. Application of these equations in their general form is complicated in that (1) the various parameters cannot easily be described in real systems and (2) the solution of these equations may require specialized mathematical skills. It should also be remembered that the simplest of electronic equipment has many systems of this sort.

Hence, the next usual approach is problem simplification. For example, it can be assumed that the systems are linear, that the spring, damping and disturbing forces follow simple and definable laws, and that the system can be described by fewer than six degrees of freedom. On the basis of these assumptions, solutions of a sort can be obtained. But with many of the more complex systems, the only expedient way to find the error in the resulting equations is to build the system and test it extensively, and this is what we wish to avoid.

Similar problem areas exist for shock and temperature for which, because of their transient nature, the methods of analysis are considerably more complicated.

Although satisfactory hardware has been produced, the problem still remains, in many cases, of adequately describing mathematically system environmental responses. If it is not generally and easily possible to develop and/or modify equations to describe existing package techniques, it might be possible to modify the packaging technique itself to achieve a solution. With this in mind, it would seem that a desirable packaging technique should provide positive but variable control over the various environments. Variability is required for the selection of the least damaging environmental situation for best performance in the total environment (the total of all the environments imposed simultaneously). Performance should be predictable with resonable degrees of precision by simple relationships in order to extend the usefulness of the technique to a wider user population. Further, control of the individual environmental factors must exist with relative independence of each other for system flexibility.

Hence, if all of these characteristics could be built into the package, development costs would be reduced because of (1) fewer engineering man-hours to establish the package design and (2) reduced environmental testing for ascertaining package responses.

4.2 Reduction of Production Costs by Use of Less Environmentally Rugged Components

The areas contributing the most to the final cost of production hardware are:

(1) Purchased commercial electronic component parts (tubes, relays, etc).

(2) Fabrication of nonprocurable parts.

(3) Assembly labor.

(4) Qualification and acceptance testing.

The costs of items (2), (3), and (4) are associated with the general excellence of the design, and may be reduced through astute engineering practices. By way of illustration, standardized modularization generally reduces the costs of fabrication because of the multiplicity of identical shop operations. Furthermore, modules may accelerate assembly and test operations by permitting parallel personnel efforts. But the attainment of environmental goals is not based on the use of modules per se. If, for example, the vibration environment is so severe that isolation techniques are obligatory, the success of the design does not depend on the modules although they may lower the total cost of the produced hardware, but on the isolation technique which lowered stresses to tolerable values.

Conversely, item 1, purchased commercial electronic component parts, does depend on the particular approaches used to satisfy environmental goals, since little can be done to modify the individual environmental characterics except in their manner of mounting. Equipment designed stiffly to withstand vibration requires more rugged components than isolated equipment. Hence, there is a decided cost advantage in using packaging methods that reduce environmental stresses. Such reductions permit greater use of less special and more off-the-shelf components that do not have to be ruggedized.

The procurement costs of commercial components generally represent a significant portion of the total cost of the final item -- in the case of fuze E in figure 1, 42 percent.* The costs of these components were high because each had to be sufficiently rugged to withstand vibration of 20 to 1500 cps without isolation.

In summarizing, production costs can be lowered by utilizing packaging that reduces environmental stresses below those that produce failure in the less special, off-the-shelf items. If this is possible, it either precludes the expense of component ruggedization programs or the premium costs of ruggedized components.

5. PARAMETER SELECTION FOR COST AND ENVIRONMENT

In conjunction with these attempts to reduce costs, the ability of the package to withstand its environment must not be lessened. The selection of the final performance responses of the package in its total environment must be made on the basis of its performance under individually applied environmental stresses. These individual responses will, therefore, be considered.

5.1 Structural Vibration

In guided missile packaging, usually the most difficult environment to be contended with is structural vibration, i.e., vibrations varying in frequency from 10 to 2000 cps (table 1). The figure of 2000 cps was selected as a round number because at one time this was a practical upper value of vibration test equipment. With the many individual resonances of components (relay, vacuum tubes, etc) occurring from several hundred through 2000 cps, there is a decided advantage in isolating the vibrations from the package. In isolated equipment, isolation of the environmental vibrations takes place above the resonant frequency of the mount and the higher the exciting frequencies lie above that of the mount resonance, the higher are the degrees of isolation usually expected. This is not completely true for real equipment that exhibits large amounts of nonrigidity, but isolation generally

* Included in this percentage were several purchased precision castings. The costs of the electronic component parts were somewhat less than this value.

makes the job easier for the designer (although it adds to the difficulties of the missile overall design). The problem, then, is one of selecting the isolator resonance. This is considered on the basis of the broad range of frequencies encountered in guided missile applications (10 to 2000 cps or higher) and apart from the availability of commercial isolators.

One basic factor in the design of isolated packages is the displacement of the equipment at resonance, which, in guided missiles where space is at a premium must be minimized. Isolator designs generally attempt to control package transmissibilities to 2-3:1 or lower. It appears that 2-3:1 is also a respectable transmissibility in stiff packages. For example, the fuze shown in figure 1 had 4-5:1 transmissibilities in the region 1000 to 2000 cps and was able to survive acceptance tests satisfactorily for 15-g inputs. For the 15-g inputs suggested in table 1 and 2-3:1 transmissibilities, figure 3 shows that the package (or mass) displacements start to minimize for frequencies above approximately 100 cps. Hence, for the unique characteristics related to guided missile applications (broad frequency ranges, large accelerational excitations, and premium space), it would seem that isolator frequencies above 100 cps are desirable from a displacement viewpoint.

Below approximately 30 cps, most tests limit the amplitudes of the exciting vibrations, and it is in this region that most commercial isolators resonate. However, the package displacements at these frequencies for 2-3:1 transmissibilities are fairly large and the ability of the isolators to absorb shocks is usually limited to fairly low values (15 to 30 g or less). Hard bottoming is frequent. The amplification of shock experienced by the package due to the shock pulse shapes and durations are not usually considered.

While displacements of the isolated package are minimized for frequencies above 100 cps, there are reasons for keeping the resonance as low as possible. Ruzicka and Cavanaugh (ref 8), in a study of the effects of nonrigid bodies on vibration isolation, point out that the assumption of a rigid body for isolation gives an "unconservative estimate of the true response for most vibration isolation problems." By developing a mathematical model and through experimentation they graphically portray the effect of nonrigidity on the transmissibility of a simplified system (an undamped free-free or unrestrained beam supported on a linear massless spring). This was done for the resonant frequencies of the beam 2, 5, and 10 times higher than the natural frequency of the spring (i.e., the mount frequency). Figures 4, 5, and 6 reproduced by

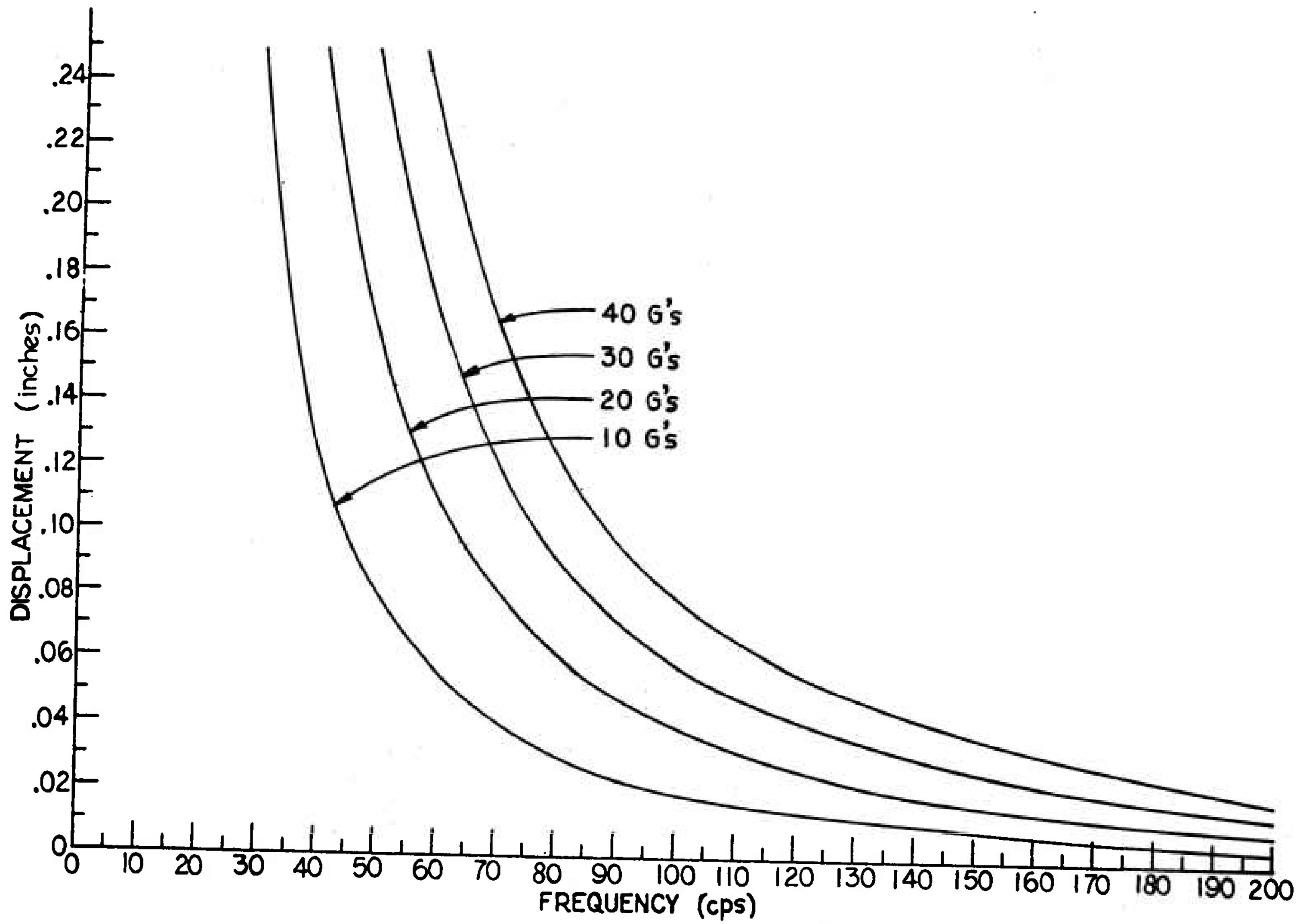

Figure 3. Mass displacements for constant mass accelerations

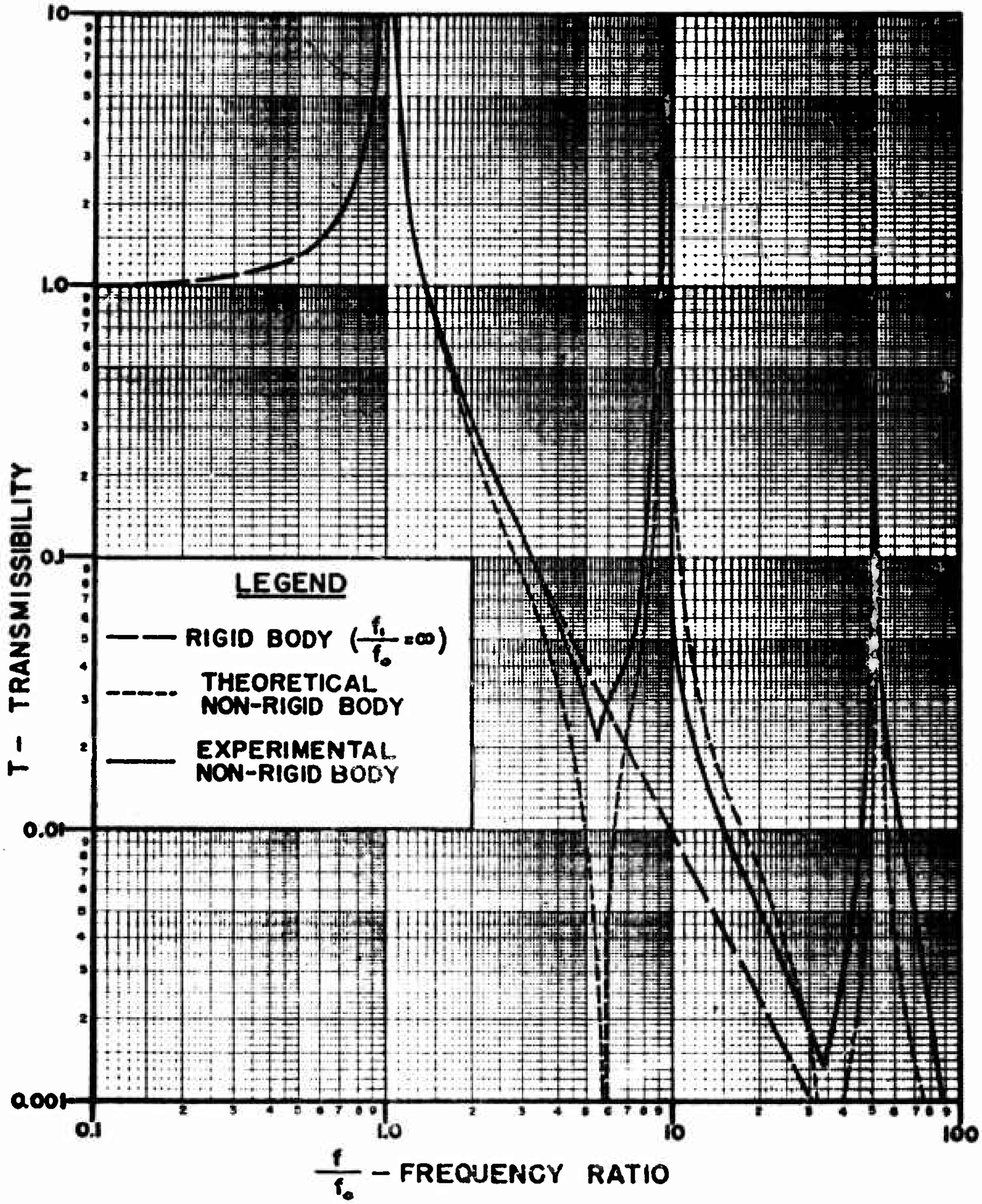

Figure 4. Transmissibility for $f_1/f_o = 10$

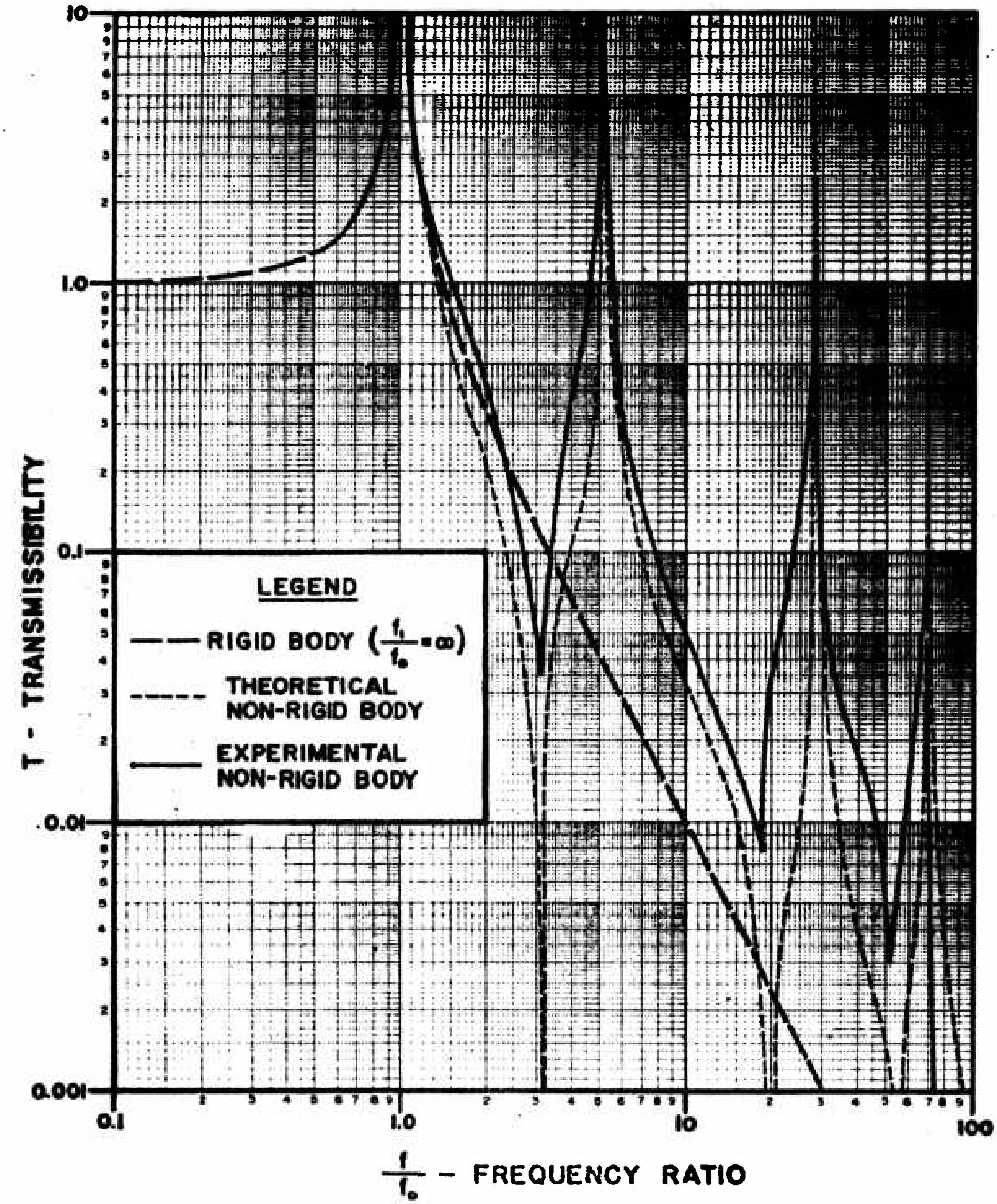

Figure 5. Transmissibility for $f_1/f_o = 5$

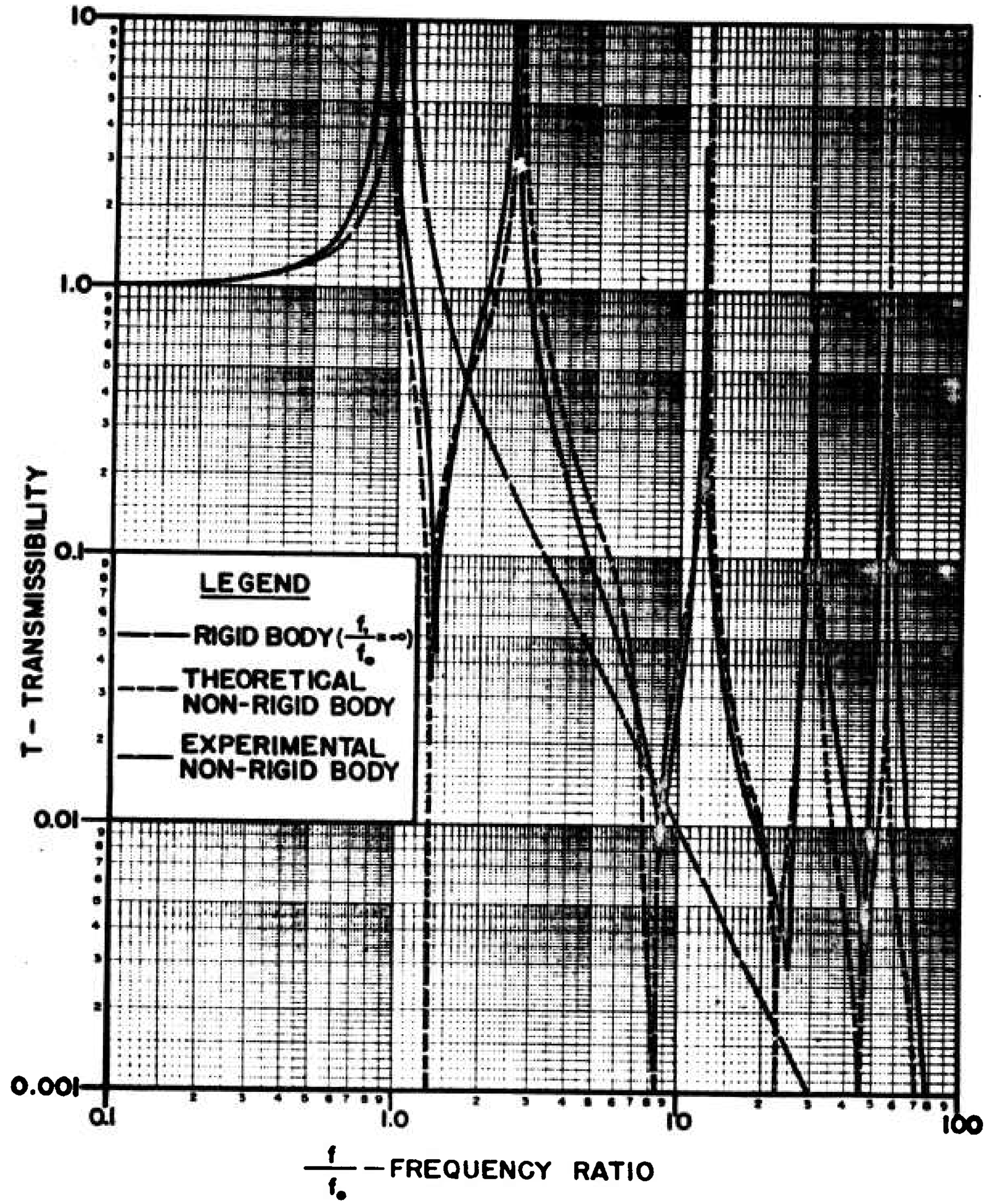

Figure 6. Transmissibility for $f_1/f_0 = 2$

permission from reference 8 are the resulting transmissibility curves for the rigid body, the theoretical nonrigid body, and the experimental nonrigid body. The forcing frequency is defined as f, the resonance of the free-free beam is f_1, and the natural frequency of the rigid body on the spring is f_o. It should be observed that the experimental and calculated responses do not completely correspond.

Three important benefits occur for large values of $\frac{f_1}{f_o}$ or for high package resonances relative to the isolator resonance:

(1) The dynamic properties of the system are predictable by simple rigid-body theory over a larger frequency range. This assists in reducing the duration of development programs and, hence, reduces costs as discussed earlier.

(2) Less vibration is transmitted by the spring (or isolator) at the fundamental resonance of the beam, thereby improving transmissibility characteristics and reliability.

(3) If the beam or package is made stiffer, there are fewer resonant conditions over a specified frequency range.

The ratio $\frac{f_1}{f_o}$ may be made large by decreasing the isolator resonance or by increasing the package resonance. Ruzicka and Cavanaugh (ref 8) point out that softening the isolator tends to "lower all the peaks of the beam resonance." Within the same frequency range the resonances remain the same. Making the beam stiffer "tends to reduce the number of resonances in the same frequency band in addition to lowering their peak value."

In real equipments mounted on isolators it is, therefore, important to consider the relative stiffness of the package with respect to that of the isolator. Since the designer has some control over the package structural stiffnesses, within the bounds of space, weight, materials, etc, but much less control over component resonances, mount resonances are usually kept low. This results in improved predictability and less costly development programs by a lowering of peak accelerations in the isolated regions, those regions of uncontrolled component resonances.

5.2 Noise (Air-Induced Vibration)

Of increasing importance in guided missile packaging is the effect of air-induced vibrations on the response of structures. The range of frequencies involved is appreciable, 150 to 9600 cps (table 1) and the phenomenon must be considered in equipment intended for use in both high-performance aircraft with comparatively low velocities and in high Mach-number missiles.

C. E. Crede (ref 5, p 148) states that "isolation of noise has much in common with isolation of vibration. Noise isolation involves essentially the reduction of the force that causes a noise radiating structure to vibrate. If the force is reduced, vibration amplitude of the structure is reduced. It may be said in a qualitative sense that the application of isolators having low-force transmissibility tends to effect reduction in the noise level."

For example, the interposition of isolators between a vibrating machine and the floor produces a quieting effect on the room beneath because the driving force (of the vibrating machine) is not transmitted to the floor, which in itself would be forced into vibration, with a rise in room noise level.

The normal textbook transmissibility curve as Crede points out is based on the assumption that the vibrations are of relatively low frequency and their wavelengths are great compared with the thickness of the isolation or resilient materials. If, however, "the thickness of the resilient material is equal to an integral number of half wavelengths of the vibration, standing-wave resonances tend to develop in the resilient material and peaks occur in the transmissibility curve." These resonances disappear completely at the higher frequencies. "The resilient material then functions as a dissipative medium and transmissibility continues to decrease at a faster rate than simple theory indicates.

"Transmissibility tests of many mounts at relatively high frequencies reveal that the standing-wave resonances ... occur almost inevitably. A reference to the dimensions of commercially available isolators indicates that standing-wave resonances need not be anticipated below about 400 cps. This frequency is well within the range of audible frequencies. Standing-wave resonances must, therefore, be considered a factor in employing isolators for the reduction of noise."

Crede suggests two ways of improving a situation in which standing-wave resonances are troublesome. The first is to lower the entire fundamental transmissibility curve by lowering the fundamental natural frequency of the system. Hence, by lowering the fundamental isolator frequency, noise isolation may be increased even though stand-wave resonances are not a factor. The second is to increase the damping capacity of the isolation material which reduces the amplitudes of the standing-wave resonances. For example, metal springs have peak amplitudes of standing-wave resonances much greater than those for rubber springs because of less damping.

With reference to the customary packaging techniques, equipments using vibration isolators have a measure of protection from noise environments. Equipments designed stiffly have less protection because of the small amounts of damping available and the higher transmissibility curves.

5.3 Shock

Appendix A develops the fundamentals of the shock response characteristics of systems with variable mount resonances; these qualities are used in examining the shock response characteristics for guided missile fuzes.

To examine the response characteristics of systems with resonances forced to occur at frequencies from 20 to 2000 cps, it is necessary to define a pulse shape and a pulse duration. The dynamic load factors can then be determined. For example, if it is determined that the expected shocks are sinusoidal and of 10-msec duration (table 1) and if the system is resonant at 1000 cps, the time ratio of figure C in Appendix A is $t_1/T = .010/1/1000 = .010\ (1000) = 10$, with a corresponding load factor of about 1. Repeating this for each frequency, the primary spectrum curve (the curve showing the initial system response to shock without considering its reverberational characteristics) in figure 7 results. This shows that systems with resonances of about 250 cps or higher experience without amplification the imposed shocks. Lower resonances cause amplification or isolation. Therefore, from 250 cps up there is no advantage from a shock standpoint (for 10-msec shocks of these shapes) in designing for stiffness and, of course, considerable disadvantage from a vibration viewpoint. The introduction of damping into the system, as shown later, lowers this 250-cps limit somewhat.

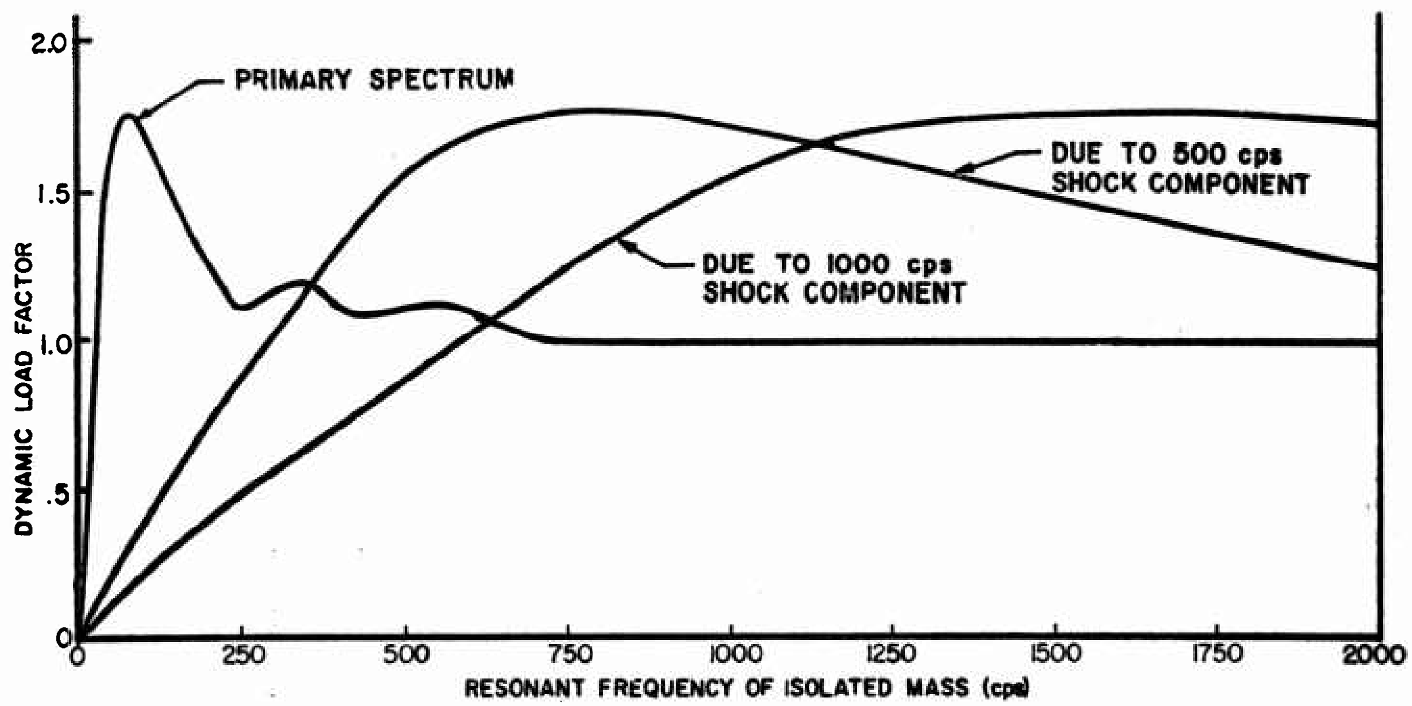

Figure 7. Locating the mount resonance of a vibration isolated electronic package to consider shock response

Figure 8 illustrates the dynamic responses of elastic systems to several other pulse shapes. The curves are reproduced, by permission, from reference 9, which discusses in considerable detail the initial response or the primary spectrum of the system resulting from the shock together with its reverberational response (residual response).

From the preceding example for sinusoidal shocks and figure 8, it is suggested that, because of the unpredictable character and duration of the shocks encountered in service, time ratios below 2 should be avoided. And to repeat, there is no advantage to design with unnecessarily large ratios, either from a shock or vibration standpoint.

Up to this point, ideal pulse shapes have been discussed. However, real shocks introduce other considerations. Figure 9 shows oscillographic records of a shock measured on a model when dropped on a steel plate. The high-frequency shock components, which can be seen superimposed on the input pulse, must be considered in the design of equipment.

Since real systems do not usually have a single resonant freqeuncy but multiple resonances (tube filaments , relays, etc), these resonances can be excited with amplification if the frequency components of the shocks are proper. For example, consider a tube with a filament resonant at 1000 cps mounted on a structure, which in itself is resonant above 2000 cps. The dynamic load factor for the structure under a 10-msec sinusoidal pulse is essentially that of the primary spectrum curve of figure 7. If the tube is shocked and if the shock has a 500-cps component, the corresponding time ratio is $t_1/T = (1/2)(1/500/1/1000) = 1$. From figure C in Appendix A the dynamic load factor for the filament is seen to be 1.75, which means the filament experiences approximately 75 percent more stress than the corresponding case of static loading. The curve denoted "500-cps component" in figure 7 demonstrates the variation of load factor for a shock with a 500-cps component. The curve shows that, for filament resonances greater than 300 cps excited by a 500-cps shock component, amplification of the shock results. Figure 7 also shows the dynamic load factors for systems with resonances to 2000 cps when subjected to shocks with a 1000-cps component. According to this curve, shock amplification occurs for frequencies greater than 600 cps.

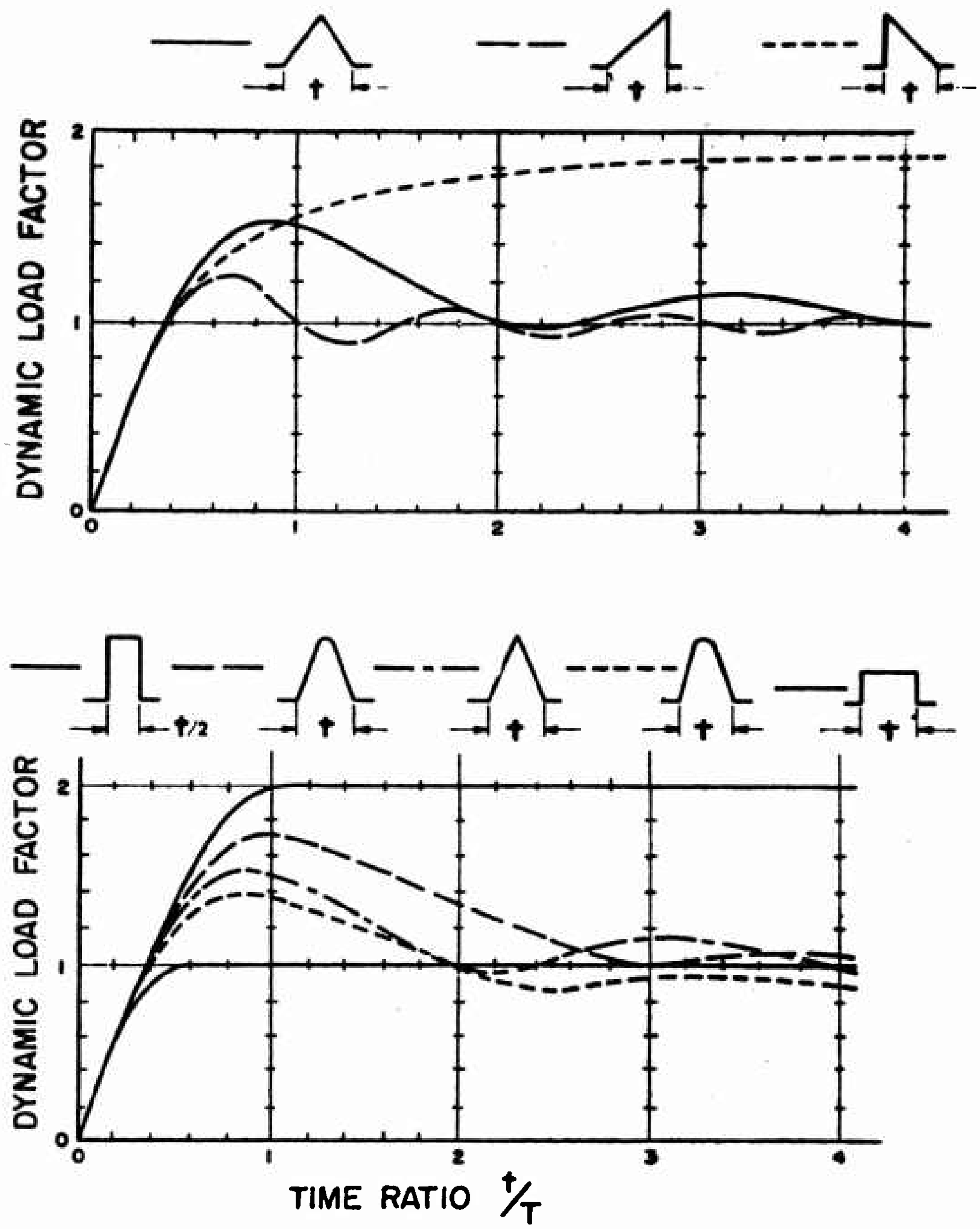

Figure 8. Dynamic load factor for various pulses having same velocity change

(3" DROP ON STEEL PLATE)

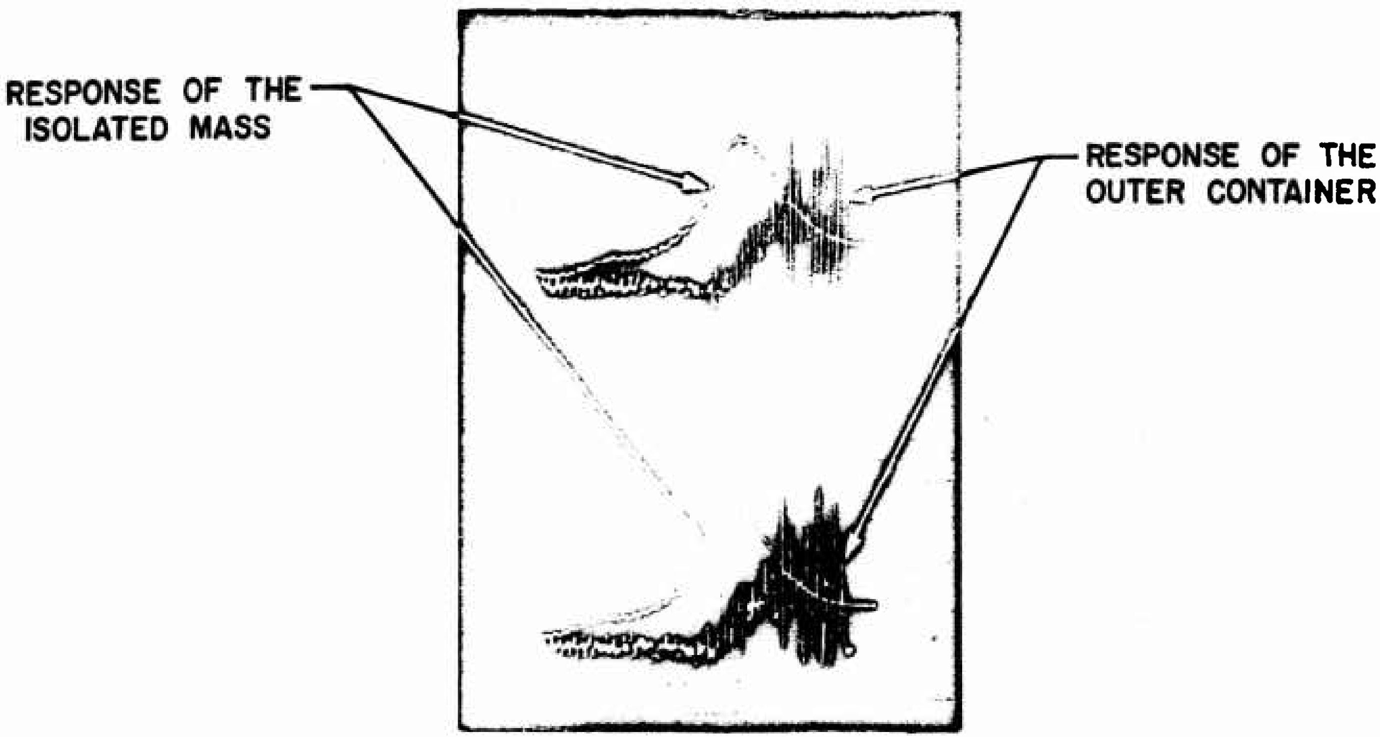

Figure 9. Sample oscillographic record of the system's shock response

The conclusion is that for real equipments with many uncontrolled component resonances and for real shocks modulated by high-frequency shock components considerable benefit would be achieved by the filtering afforded by a vibration isolation mounting. By filtering the higher-frequency components from the shock, the many resonating components would be expected to experience the shocks without amplification. This, it seems, produces a desirable measure of control. The designer then need concern himself only with the shock response of the isolated mass.

5.4 Sustained Acceleration

The situations in missile performance imposing sustained accelerations on isolation-mounted equipment have the effect of increasing the weight of the suspended mass. Deflections are increased and if they are great enough, hard bottoming may occur. If nonlinear mounts are used and if the mount resonance is high, hard bottoming is precluded in a practical sense. The effect of sustained accelerations is to reduce the resonant frequency of the isolated mass.

5.5 Final Parameter Selection

From the preceding discussion it can be seen that it is impossible to select all package design parameters so that benefits always result. The package characteristics that would improve equipment performance under vibration are detrimental to shock and so forth. But design parameters can be selected that not only cause the equipment to meet the engineering boundaries but also simultaneously reduce costs by shortening development programs and by utilizing existing components without further ruggedization.

To achieve these goals isolation is definitely required because of the two following reasons:

(1) Many essential electronic components resonate uncontrollably below 2000 cps (but fortunately most above 500 cps), and if these are to be employed successfully, isolation is essential. Both cost and reliability are affected.

(2) Performance predictability is not easily achieved without isolation. Isolated packages perform in accordance with more simple relationships than do stiff packages, and the system constants are usually known.

The package, therefore, must be isolated. To minimize shock stresses, it must have its fundamental mount resonance at or above 250 cps (par 5.3). This also has the added benefit of minimizing package excursions at resonance (fig. 3). To keep the $\frac{f_1}{f_o}$ ratios large for performance predictability (par 5.1) the mount resonance must not greatly exceed 250 cps. For simplicity in calculation, the deflection of the mount itself should be linear over a sufficient range to handle the 50-g design shocks (table 1) with a subsequent nonlinear region to compensate for the larger unpredicted shocks without hard bottoming. Enough damping must be incorporated in the mount to prevent transmissibilities from exceeding 3:1. The thickness of the isolator should be kept small to extend the noise-free regions.

Since this study was restricted to small guided missile fuze packaging, it is possible to require structural resonances above 2000 cps. This, incidentally, is not especially difficult. The fuze shown in figure 1 had structural resonances well above 2000 cps. This requirement helps establish large $\frac{f_1}{f_o}$ ratios, hence, better performance prediction. For improvement in noise isolation, the entire structure containing the electronic component parts should be surrounded by the isolation medium.

5.6 Need for a Model

If an isolation method can be developed that exhibits these characteristics, a less costly method of packaging small guided missile fuze electronics would be available. For example, fuze C shown in figure 1 did not exhibit severe resonances below 350 to 400 cps. If isolation could have been realized below these resonances, subsequent repackaging would have been unnecessary. With an isolation system having a mount resonance at 200 to 250 cps, isolation begins around 300 cps or so, above which accelerations are reduced. Such a system would produce the benefits of stiff packaging with the added advantage of isolation at the critical higher frequencies. For the purposes of proving this, an electronic fuze need not be built because:

(1) Stiff packaging of fuzes has been successfully accomplished in the past.

(2) Isolation packaging emphasizing the ideas expressed in this report would considerably reduce the environmental stresses experienced by stiff designs.

A model must be built however, to demonstrate the practicality and feasibility of the method set forth herein.

6. MODEL DEVELOPMENT AND TESTING

6.1 Description of Model

A model was constructed as shown in figure 10, which illustrates the previous conclusions. Two aluminum cylinders, separated by angle-shaped rings of a spring material, provide control over the resonant characteristics of the inner isolated cylinder. By varying the spring stiffness, the resonant characteristic of the inner cylinder can be changed. Between the two cylinder is a damping material of cellular structure to control the excursions (displacements) of the inner cylinder at resonance, since the amount of damping in the spring is usually not sufficient by itself. To improve the volumetric efficiency of the isolation method, a cellular damping material was used that could hold an evaporative coolant. By using simple evaporation with the control media for vibration and shock, the package can be surrounded with a variable and predictable maximum temperature boundary to externally generated heat. For internally generated heat, the same boundary assumes the role of a controlled maximum temperature heat sink, or reservoir. Hence, such an approach provides within the same medium a means for controlling the dynamic forces as well as temperature, with a considerable saving in space and weight.

The spring in the model was silicone rubber because of its desirable high-temperature qualities and load deflection curve. Polyurethane foam was chosen for the damping material. This material possessed satisfactory damping characteristics, and it had the ability to absorb and hold a fluid. While it deteriorated at temperatures above 300°F, the coolant evaporating at lower temperatures maintained its character.

To raise the noise-free region at the lower end of the frequency spectrum, the space between the cylinders was reduced to 5/16 in. This is a considerable reduction over currently available isolator thicknesses.

Figure 11 illustrates a possible application of the isolation method to real equipment. It should be observed that the isolation medium completely surrounds the isolated electronics for greater noise reduction. With the inner cylinder resonating above 2000 cps, and mount resonances

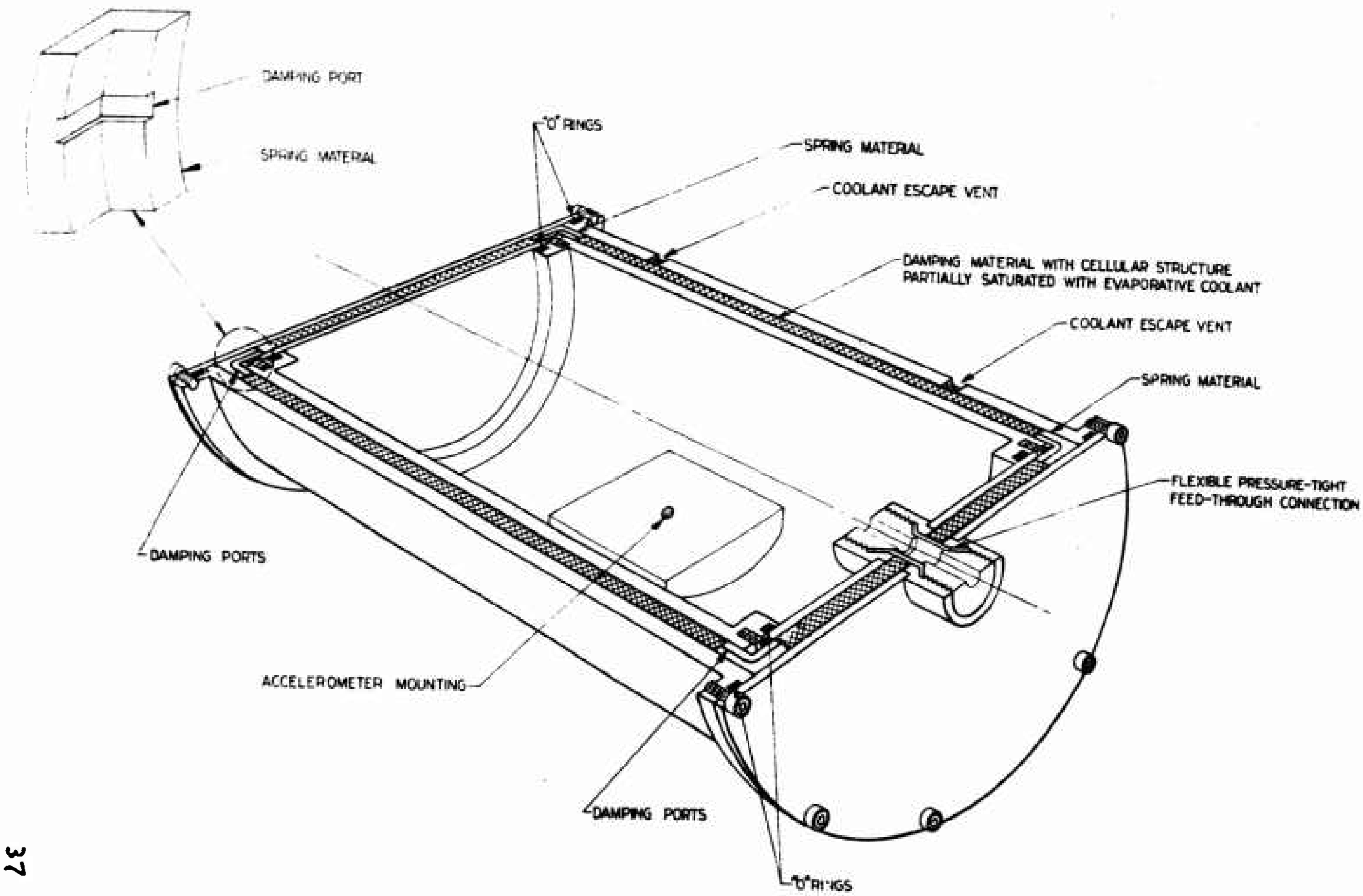

Figure 10. A model to demonstrate controlled environment responses

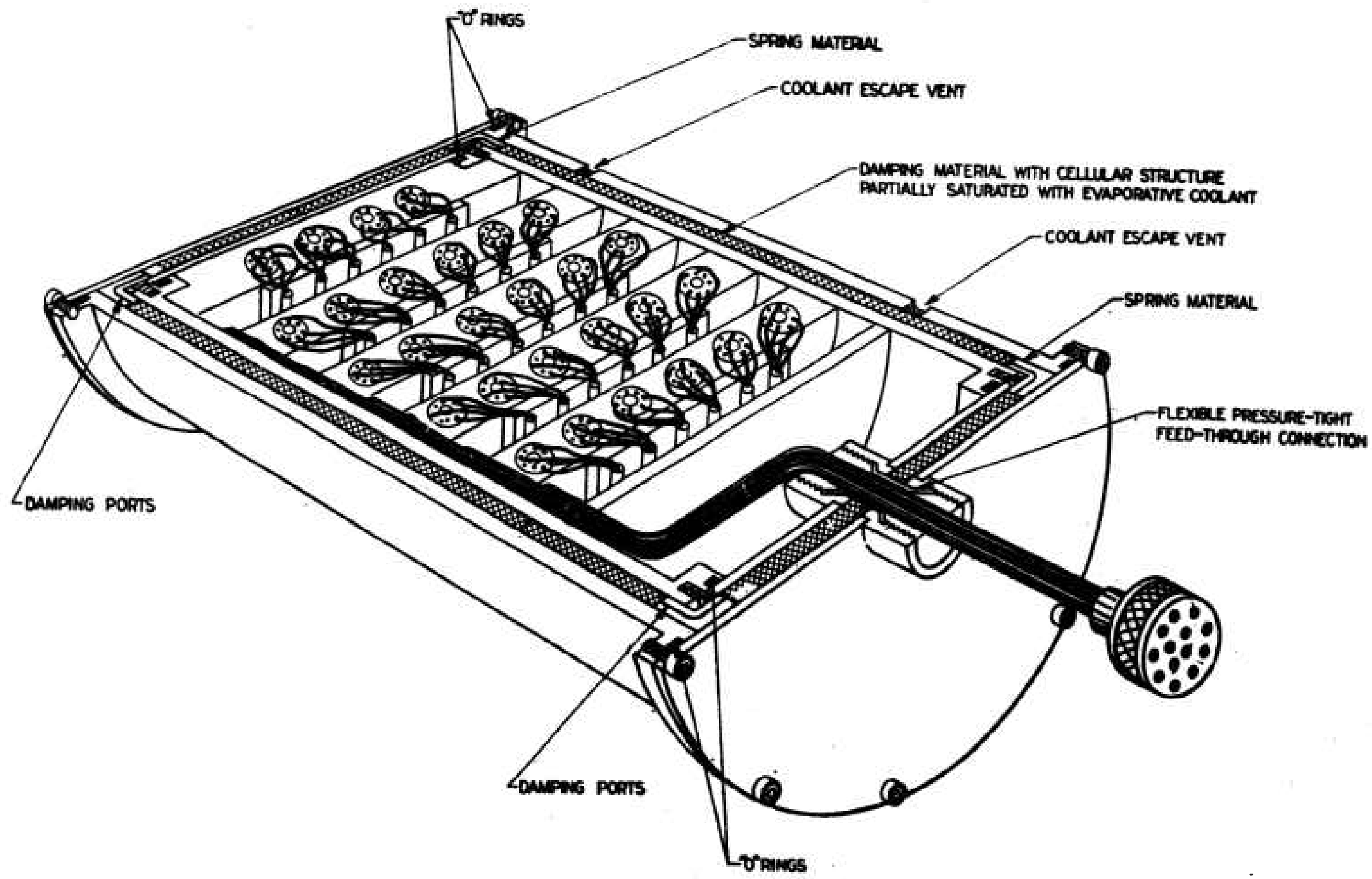

Figure 11. An application of the technique for controlling environmental responses

between 200 and 300 cps the requirements set forth in the previous section are realized.

6.2 Structural Vibration Characteristics

Figure 12 shows the response of the complete system without fluid for longitudinal and diametrical vibrations. For the longitudinal vibrations, it is observed that the mount resonance occurs below 200 cps (below the desirable limit). This could be easily corrected by a stiffer spring. It may also be seen from figure 12 that, as the levels of input vibrations increase, the transmissibilities decrease. This is particularly encouraging in controlling maximum accelerations of the isolated mass. Figure 13 shows the variations of transmissibility with input acceleration for those conditions in which the rubber rings only control the system response, and for the complete system with and without fluid saturating the damping material.

Figure 13 also demonstrates that air damping is as effective as the foamed damping material at the higher acceleration inputs for longitudinal vibration. For endwise vibrations the air is partially trapped within the rubber rings and when the volume is changed by system response, the trapped air must be forced through the damping parts. Conversely, for transverse vibrations the air does not contribute significantly to the damping force because it is not so completely trapped. Damping is achieved principally by the foam material. The fluid appears to lower the damping force somewhat, causing increases in transmissibility and upward shifts in resonant frequency.

Figure 14 presents the resulting accelerations of the isolated mass vibrating at resonance. Figure 15 shows the percentage increase in the accelerations experienced by the isolated mass as a function of percentage increase in input. It may be observed that the percentage increases of mass accelerations are considerably less than the corresponding input increases.

Figure 12 also shows a distinct shift of the resonant frequency to lower values as the input accelerations increase. Figure 16 shows this shift for the various combinations of mountings.

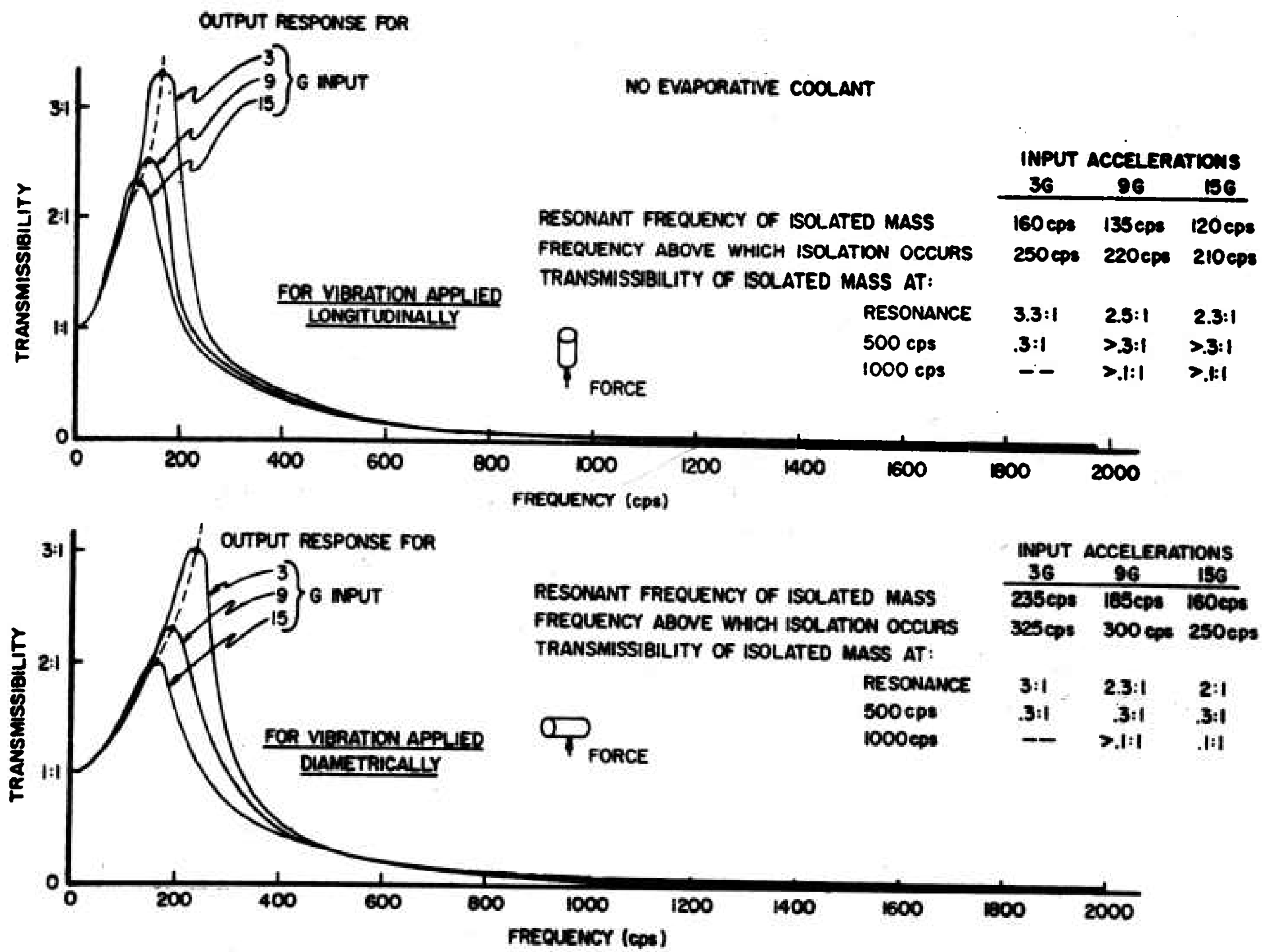

Figure 12. Vibrational response characteristic of the complete assembly

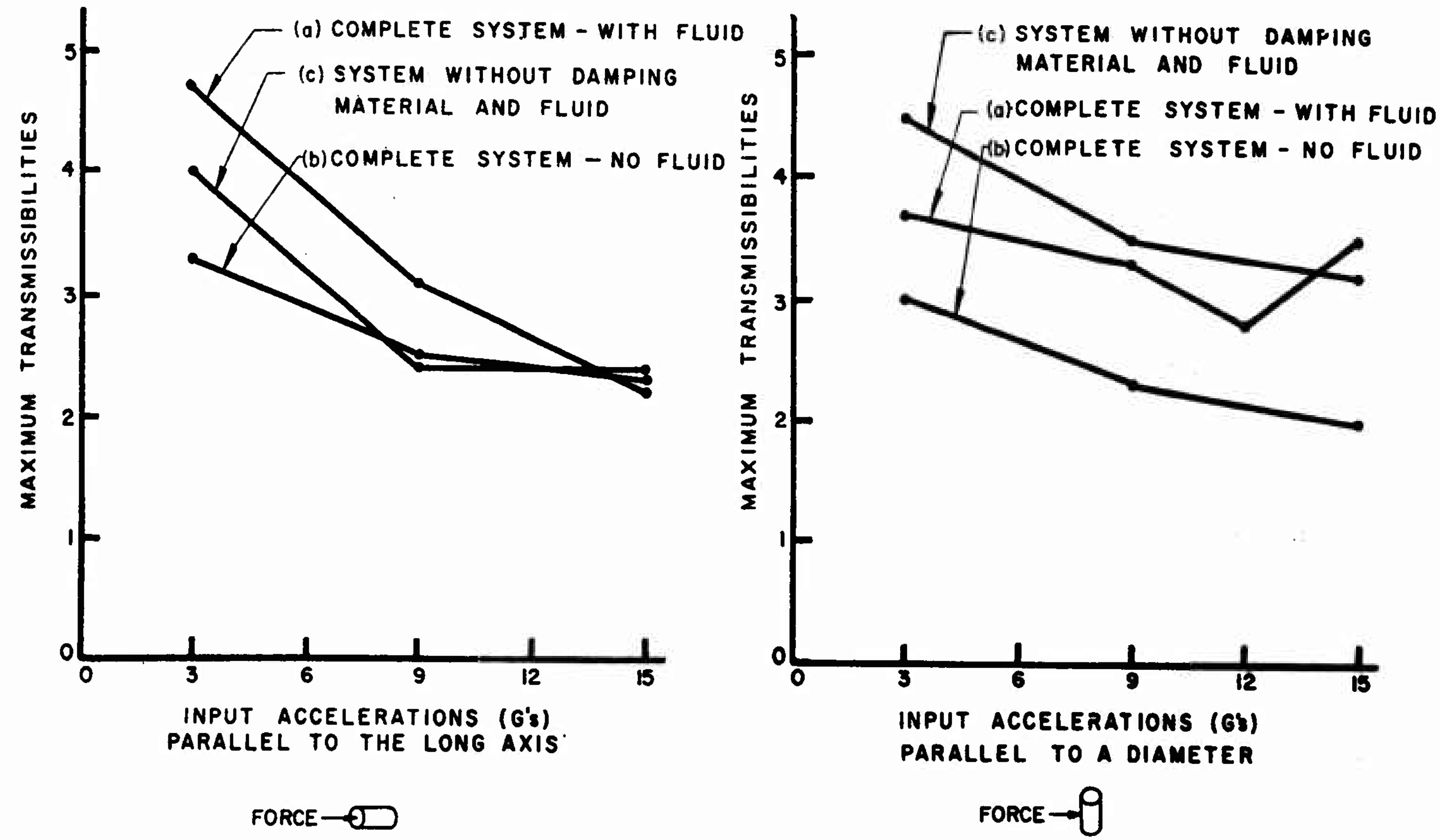

Figure 13. Transmissibilities of the isolated mass at resonance

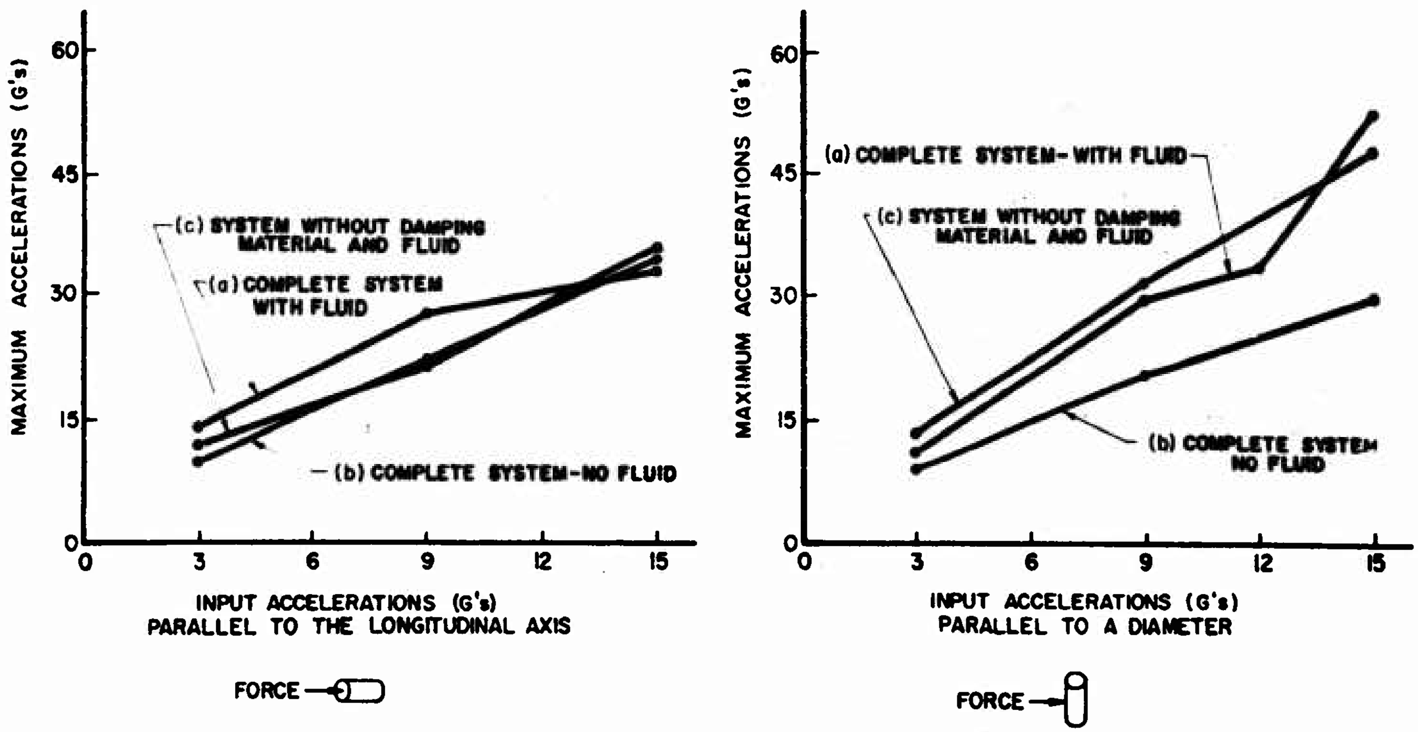

Figure 14. Accelerations of the isolated mass at resonance

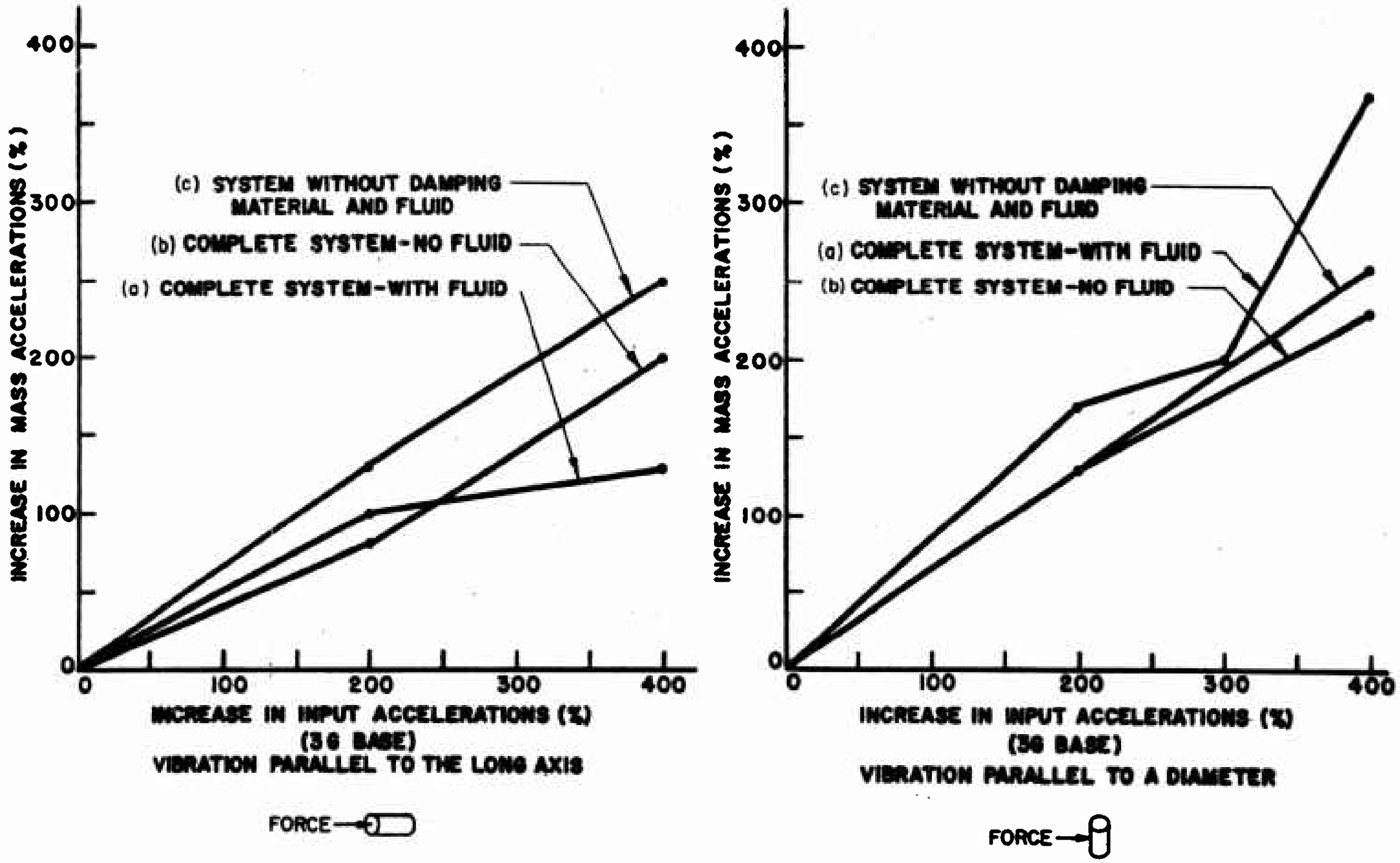

Figure 15. Increase in accelerations of the isolated mass at resonance

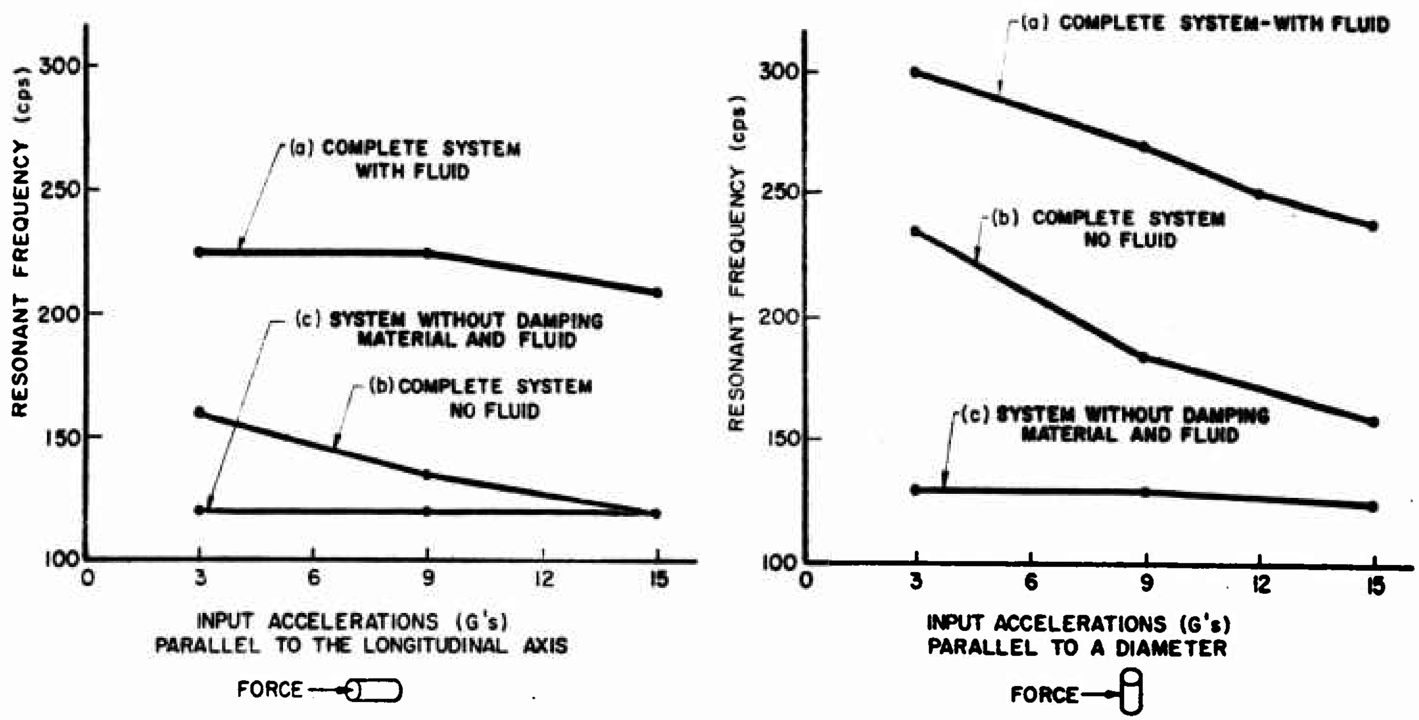

Figure 16. **Decrease in resonant frequency of the isolated mass**

The various significant values of system performance under vibration are given in table 2. Again it is observed that some resonances fall below 200 cps. These are attributable to the unavailability of materials, test facilities, etc, to correct for changes in resonances with changes in the test conditions. These low-resonant conditions however, do not destroy the demonstrative value of the model and its tests. With selection of different damping materials, these transmissibilities can be adjusted upward or downward. Rubber rings of different stiffness and size would cause corresponding changes in the resonant frequency of the siolated mass, etc. Considerable development is required, however, for an understanding of material performance, both with respect to the fundamental response functions and their character under temperature and other environmental extremes. No transmissibilities above 1000 cps are recorded because the transmissibility curves approach zero asymptotically at the higher frequencies and values near 2000 cps for the particular recording system used had little significance.

Lest the smoothness of the response curves of figure 12 lead to doubt of their validity a sample chart record is shown in figure 17. Observe that even though the input 10-g vibrations near 2000 cps could not be precisely controlled, the isolated mass and the response levels were not generally disturbed. Since the mass vibrations are approximately 180 deg out of phase with the input forcing vibrations, the mass displacements are small.

To assess the ability of the isolation materials to withstand sustained vibrations, the model without fluid was vibrated at resonance under room temperature conditions for 30 min, without deleterious effects. Total test time at all frequencies was in excess of 10 hr and with the exception of some deterioration of the foam material, no damages were noted. The system performance also remained constant with test time.

6.3 Shock Characteristics

Figure 18 demonstrates the dynamic load factors peculiar to the model for several time ratios. Because of the large amount of damping in the design, the maximum load factors are less than those for undamped systems. However, they are still of sufficient magnitude to be considered.

Table 2. System Vibration Response Characteristics

	Spring Material Only			Damping Material Only			Damping Material with Fluid			
LONGITUDINALLY	3g	9g	15g	3g	9g	15g	3g	9g	12g	15g
RESONANT FREQUENCY OF ISOLATED MASS	120	120	120	160	135	120	225	225	—	210
FREQUENCY ABOVE WHICH ISOLATION OCCURS	210	170	160	250	220	210	375	375	—	375
TRANSMISSIBILITY OF ISOLATED MASS AT:										
RESONANCE	4:1	2.4:1	2.4:1	3.3:1	2.5:1	2.3:1	4.7:1	3.1:1	—	2.2:1
500 cps	0.2:1	0.1:1	0.3:1	0.3:1	>0.3:1	0.3:1	0.4:1	0.6:1	—	0.7:1
1000 cps	—	>0.1:1	—	—	>0.1:1	>0.1:1	—	0.1:1	—	0.5:1
DIAMETRICALLY										
RESONANT FREQUENCY OF ISOLATED MASS	130	130	125	235	185	160	300	270	260	240
FREQUENCY ABOVE WHICH ISOLATION OCCURS	190	235	225	325	300	250	425	410	400	380
TRANSMISSIBILITY OF ISOLATED MASS AT:										
RESONANCE	4.5:1	3.5:1	3.2:1	3:1	2.3:1	2:1	3.7:1	3.3:1	2.8:1	3.5:1
500 cps	>0.1:1	0.1:1	0.1:1	0.3:1	0.3:1	0.3:1	0.5:1	0.6:1	0.5:1	0.5:1
1000 cps	—	>0.1:1	>0.1:1	—	>0.1:1	>0.1:1	>0.1:1	>0.1:1	>0.1:1	>0.1:1

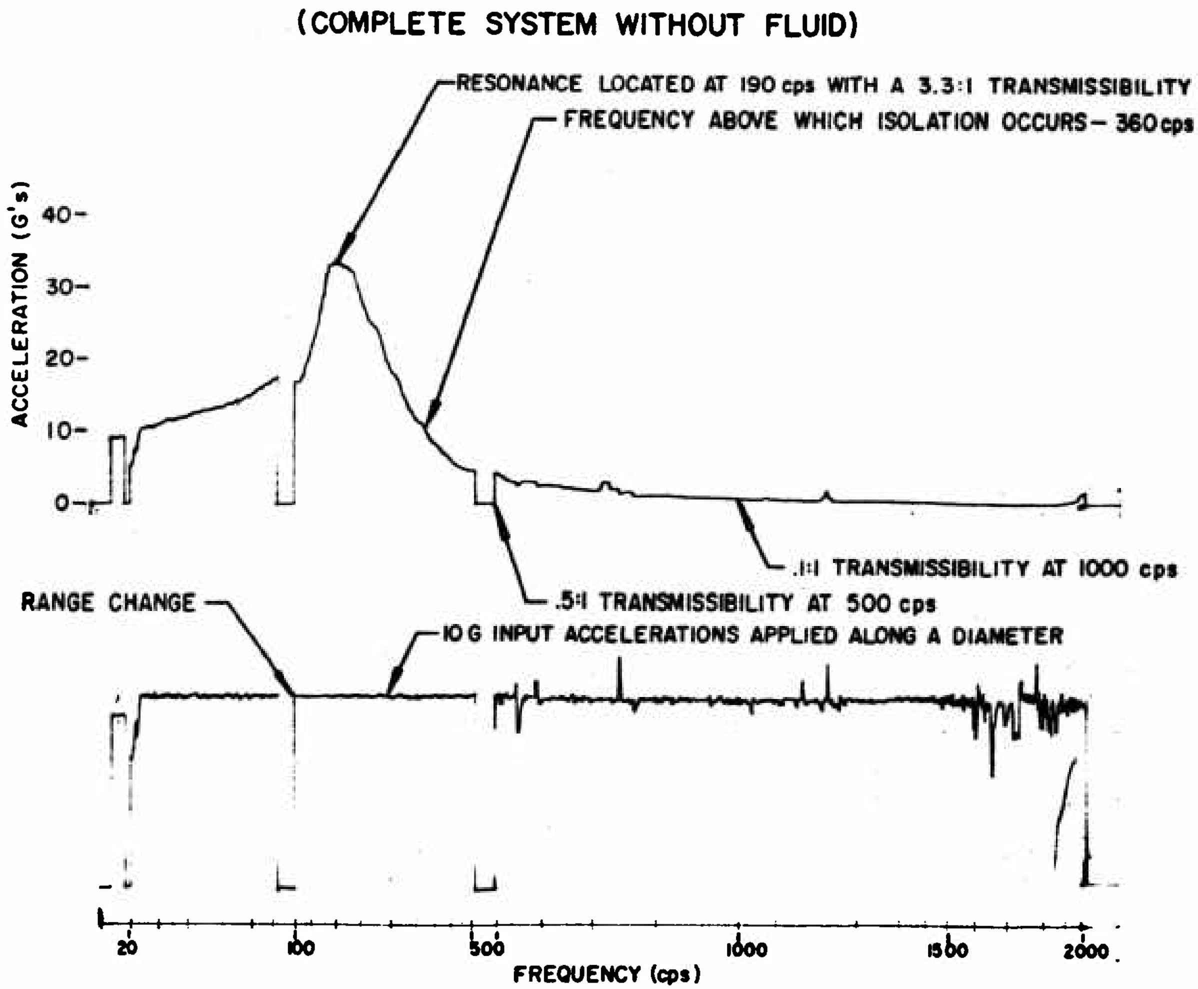

Figure 17. Sample chart record of system response

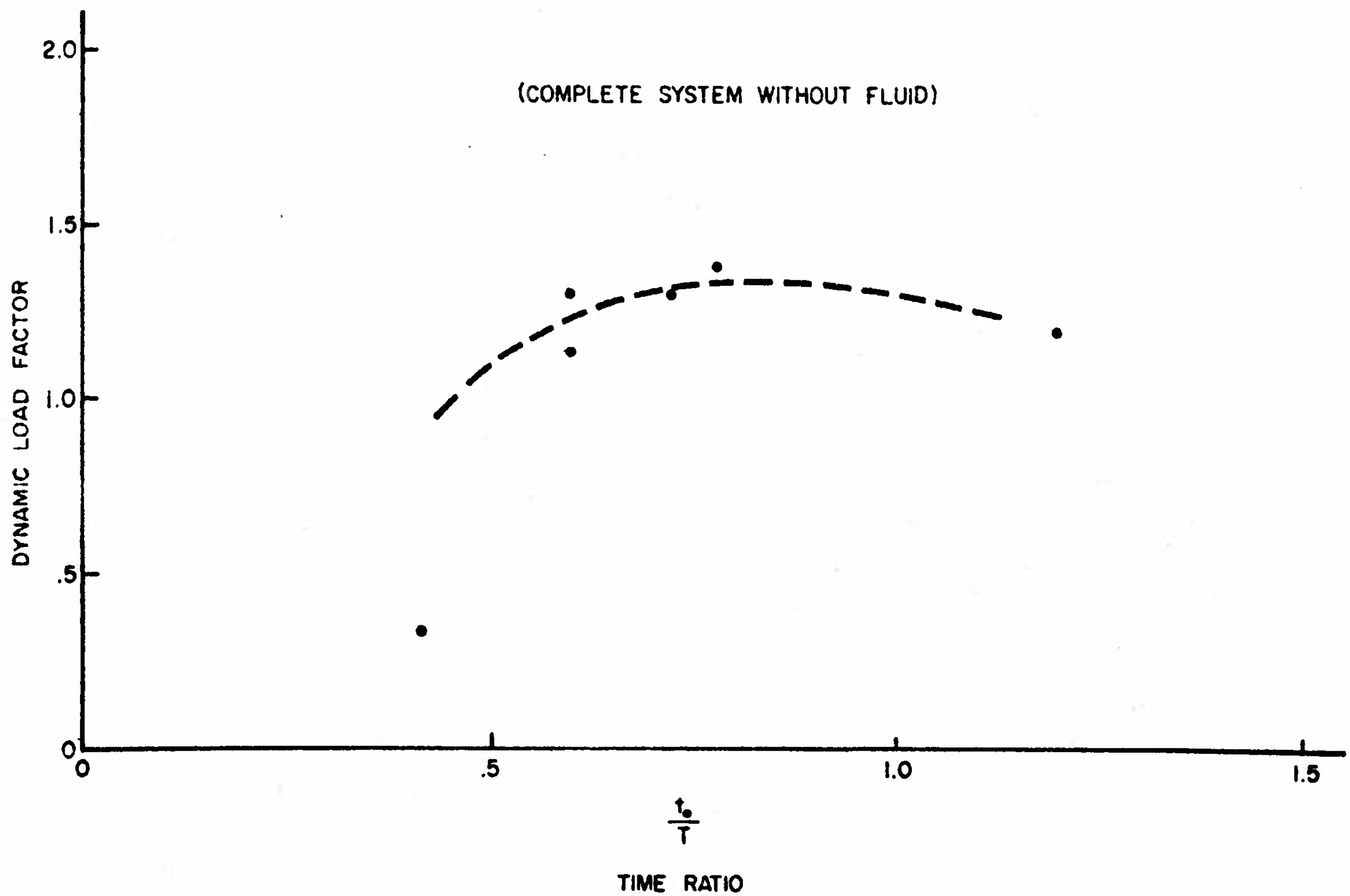

Figure 18. Dynamic load factors for sinusoidal pulses

Figure 9 demonstrates the filtering effect of the isolation medium on the impinging drop shocks.

Figure 19 shows the practical effect of dynamic loading. For the same range of drops and for the same physical hardware, the shock response of the isolated mass on rings of different stiffness is noticeably different because of the dynamic loading phenomena. The time ratios for the two series of drops fell between 0.5 and 1.5.

Finally, the pulse-shaping character of the system remains to be considered. Dynamic loading has been shown to be a function of the shape of the input pulses, as shown in figure 8. Because of the unpredictability of the shock shapes encountered in field use, the designer does not know what to expect in system response although, as pointed out in the previous paragraph, the probable best approach is to design for time ratios greater than 2. The single test performed on the model seems to suggest that this unpredictability is changed to predictability from the viewpoint of the components (tubes, relays, etc) mounted on the isolated mass, because the poorly defined pulse experienced by the exterior container in passing through the isolator is shaped into an approximate sine wave. Even though the benefits of attenuation are not realized, response predictability is -- and this is fairly important.

In summary, the two principal advantages (relative to shock) offered by a high-frequency isolation system and demonstrated in the model tests are shock filtering and possible pulse shaping. While attenuation cannot be realized either in this design or in systems designed stiffly, amplification can be prevented, which is a decided advantage. Low-frequency isolation could theoretically afford attenuation for low-g pulses, but practically it cannot because of hard bottoming characteristics which, in turn, amplify shock above the hard-bottoming threshold.

6.4 Noise Attenuation

Analysis of the model under noise environments is complicated by the fact that uniform noise fields are difficult to establish without elaborate test facilities, and the customary methods of measuring noise intensities in nonuniform noise fields (such as in the available test facilities) are highly direction sensitive. To expedite tests and make the most effective use of the available sound chamber, accelerometers were mounted on the isolated inner chamber and the

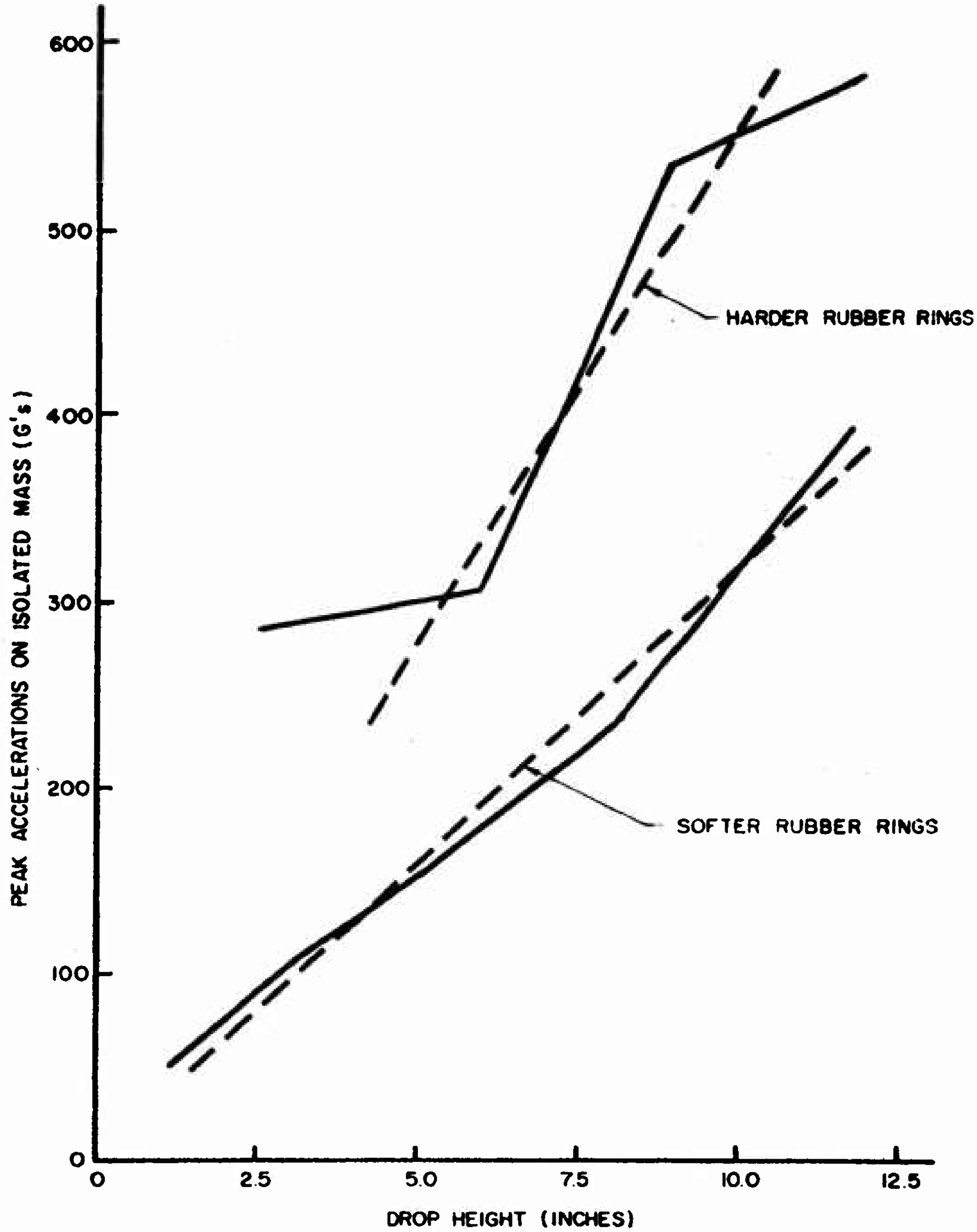

Figure 19. System response for drops on steel plate

outer shell, and measurements were made of the corresponding vibrations caused by speakers, the source of the noise. The frequency of the speakers was varied from 20 to 20,000 cps, and the resulting vibrations on the cylinders were measured together with the sound intensity level in the surrounding air. A constant voltage was applied to the speakers over the frequency spectrum. A single microphone-and-accelerometer orientation was used. Figures 20, 21, and 22 show the results of the tests.

From these records, it is concluded that considerable noise attenuation (with respect to both amplitude and frequency) is afforded the isolated mass when air is used as a void medium. Almost as much attenuation is realized when foamed plastic fills the void. Figure 20 indicates vibration attenuation through the isolation medium in excess of 10:1, and shows the noise-free area at the low end of the frequency spectrum to be extended to 1500 cps, or approximately 3 1/2 to 4 times greater than that experienced by commercial isolators. When fluid is added to the foamed plastic, the attenuating character of the system is lost. Whether or not lesser amounts of fluid would regain the attenuation qualities of the system remains to be determined. The amount of fluid used in the test was large, probably well in excess of that required for short flights.

If lesser amounts of fluid do not restore the desirable attenuation qualities for noise, the attenuating character of the system can easily be regained by separating the fluid from the isolation barrier and forming a fluid reservoir around the barrier, but not coincident with it. As a matter of fact, the fluid could be placed either around or within the isolation void with related benefits. By separating the isolation materials from the cooling fluid, vibration, shock, temperature, and noise control would be benefited considerably but with a sacrifice in weight and space.

It is possible that a sublimating substance might compensate for the disadvantages of a fluid. A powdery solid might also add to the available damping force for the isolated mass under vibration.

6.5 Sustained Acceleration

Circumstances did not permit the evaluation of the model under sustained accelerations. However, such tests were performed on a

(20 TO 20,000 CPS)

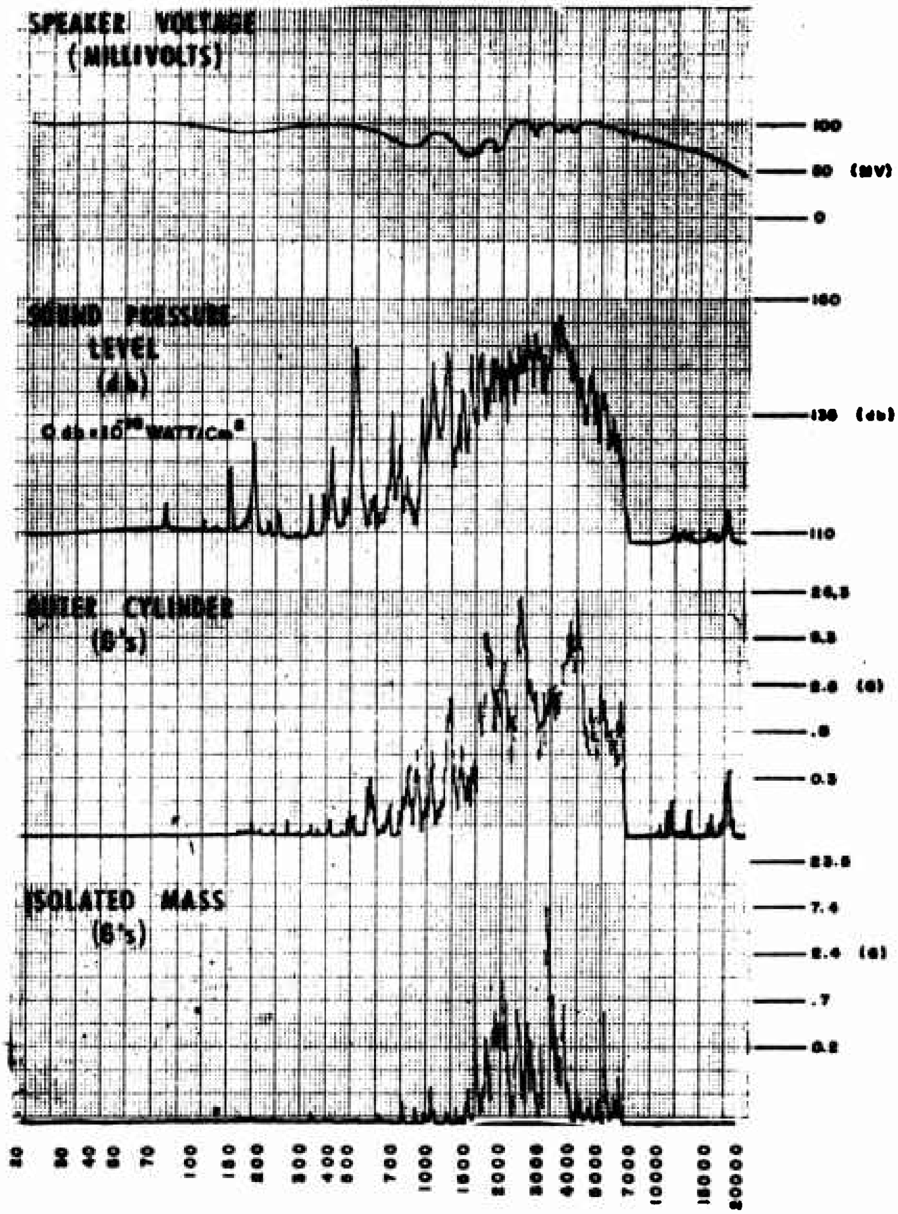

Figure 20. System noise response, rubber rings only

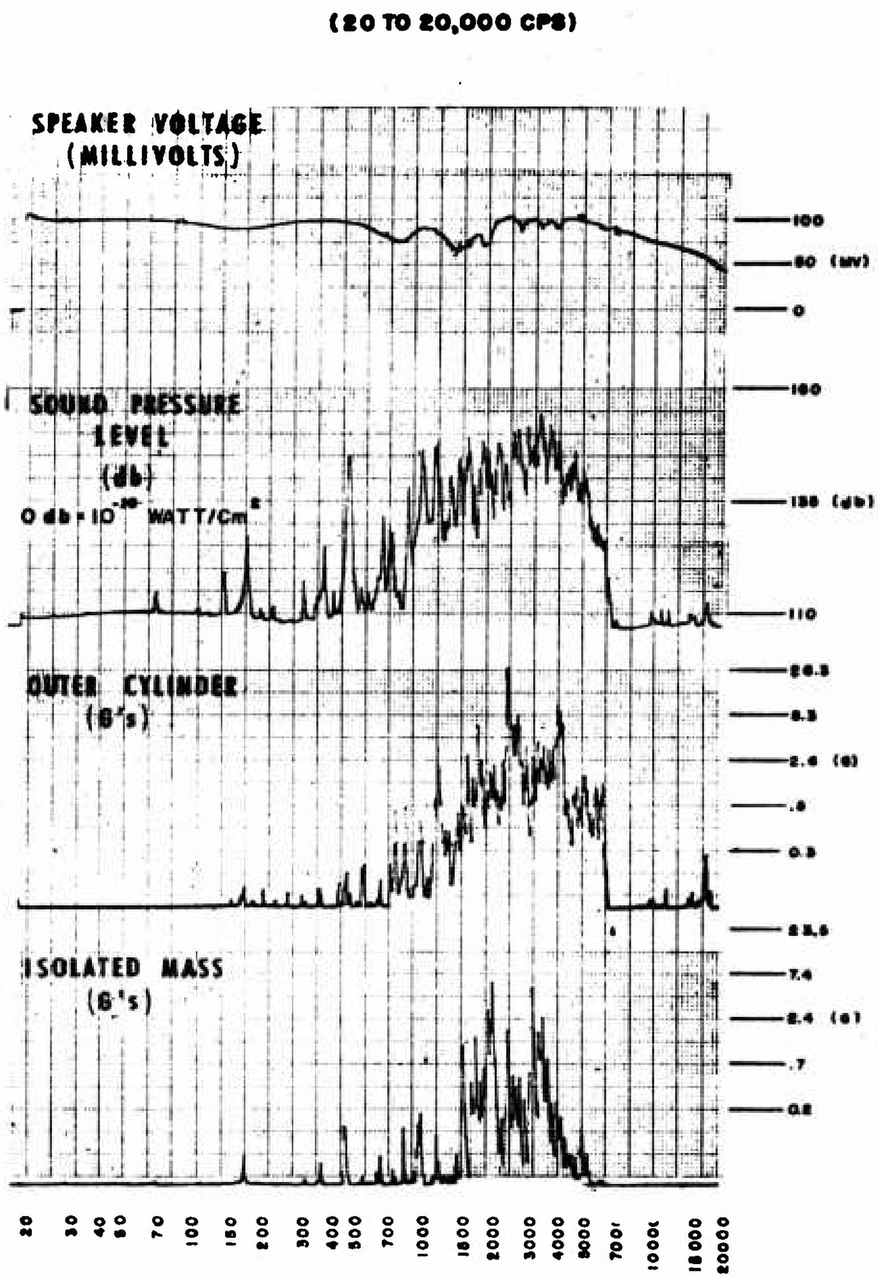

Figure 21. System noise response, rubber rings and damping material

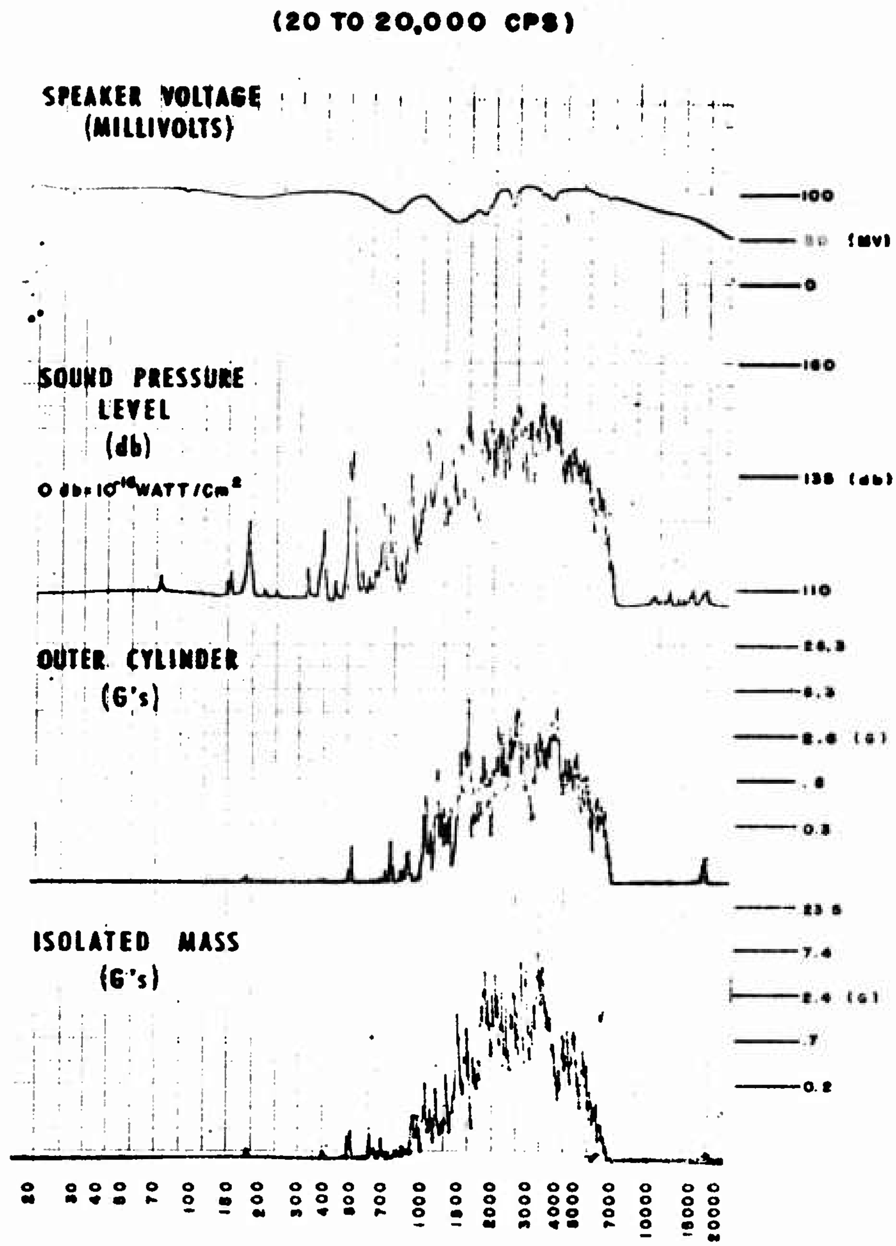

Figure 22. System noise response, rubber rings, damping material, and cooling fluid

model utilizing compression-type snubbers having a mount resonance near 80 cps. While the resonant frequency of the mass on the rubber spring was lowered, as expected, the transmissibilities were not significantly affected.

The lowering of the model mount resonances under sustained accelerational forces should not greatly affect its performance or interfere with the resonances of individual components.

6.6 Temperature Control

As mentioned earlier, the customary techniques for vibration isolation are wasteful of space. The void spaces necessary to allow for bouncing and the mounts consume large amounts of space that cannot be used for other purposes. The method employed in the model permits the void to act as a reservoir for evaporative cooling liquids, thus improving volumetric efficiency. This is accomplished by using a cellular damping material and by partially saturating the material with liquid. Because the material is only partially saturated, the vibration characteristics of the system are not greatly altered. Polyurethane foam was found to work sufficiently well to demonstrate the method.

The ability of the evaporating liquid to control the wall temperature of the inner housing is shown in figure 23. This figure compares the temperature rise of the inner chamber alone when placed in a 500°F oven (curves 1), the inner chamber when the outer chamber and foam surrounds it (curves 3), and, finally, the inner chamber when the foam is saturated 50 percent by volume of the in-between void (curve 2). The particular coolant used was a 40 to 60 percent mixture of methanol alcohol and water, because the specific heat of the mixture was roughly half that of water, and its boiling point and freezing point were 167°F and -85°F, respectively, both just beyond the military specification temperature extremes for storage. The plateau in curve 2 is caused by the alcohol boiling off, after which the curve rises to the boiling point of water and remains there.

The principal advantage of this arrangement is that packages can be surrounded with a variable and predictable maximum temperature boundary to externally generated heat. For internally generated heat, the same boundary assumes the role of a controlled maximum-temperature heat sink. Further, these very desirable advantages are achieved by making use of existing control media. The only weight added

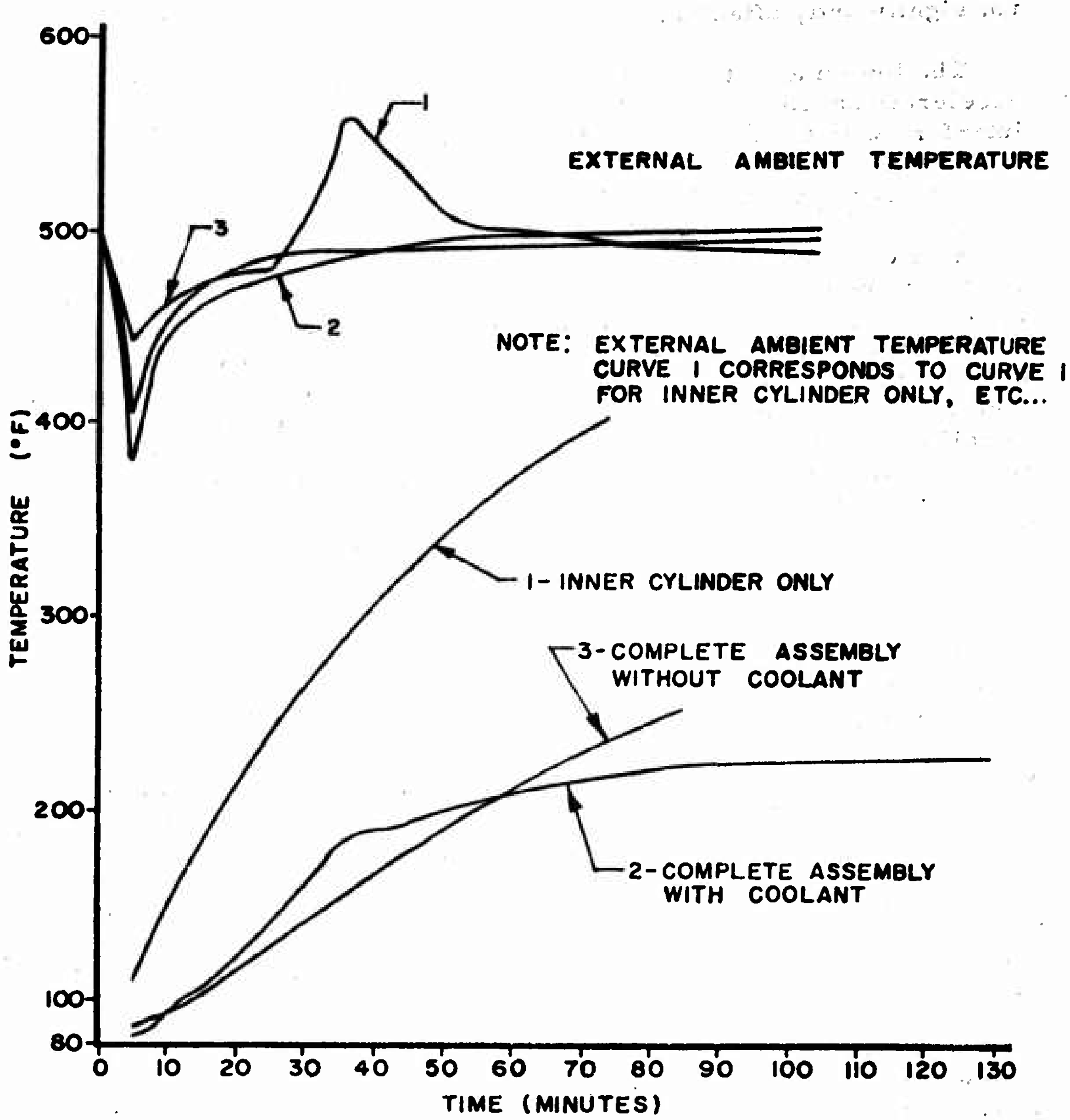

Figure 23. Comparative temperature response curves

(1 lb) was that of the fluid, and no space was required other than that already being used for control.

By using fluids of different boiling points, variable temperature control of the boundary is achieved. The specific heat of the selected fluid must be considered in conjunction with the heat inputs (both external and internal) and the missile flight durations if the maximum temperatures are not to be exceeded.

7. CONCLUSIONS

The real cost of any electronic device is not that cost required to manufacture the end product, but the cost that includes its share of of the financial burden of development and production. Consequently, to reduce real packaging costs, it is necessary to reduce any or all of those costs associated with the constituent cost factors.

Significant reductions in missile electronic packaging costs can be achieved (1) by shortening the lengthy development programs normally required for such hardware with design approaches possessing a high degree of performance prediction, and (2) by reducing environmental effects on package components to the extent that existing shelf items can be utilized to lower production costs. The report has described ways in which these might be accomplished.

It was assumed that the satisfaction of environmental and reliability objectives are essential design determinants, and that the customary approaches to packaging do not adequately consider the total environment and its relationship to costs.

Had the techniques suggested for controlling the dynamic environment been incorporated into model C of the proximity fuze shown in figure 1, the need for the development of models D and E would have been eliminated with approximately a 43-percent reduction in program costs. Obviously there are situations where such techniques would be partially or totally inappropriate.

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Appendix A

The following paragraphs present a preliminary discussion of the dynamic reaction of structures to shock pulses and are quoted from Appendix B of NRL Report 4789, "Design of Shock- and Vibration-Resistant Electronic Equipment for Shipboard Use," by H. M. Forkios and K. E. Woodward.

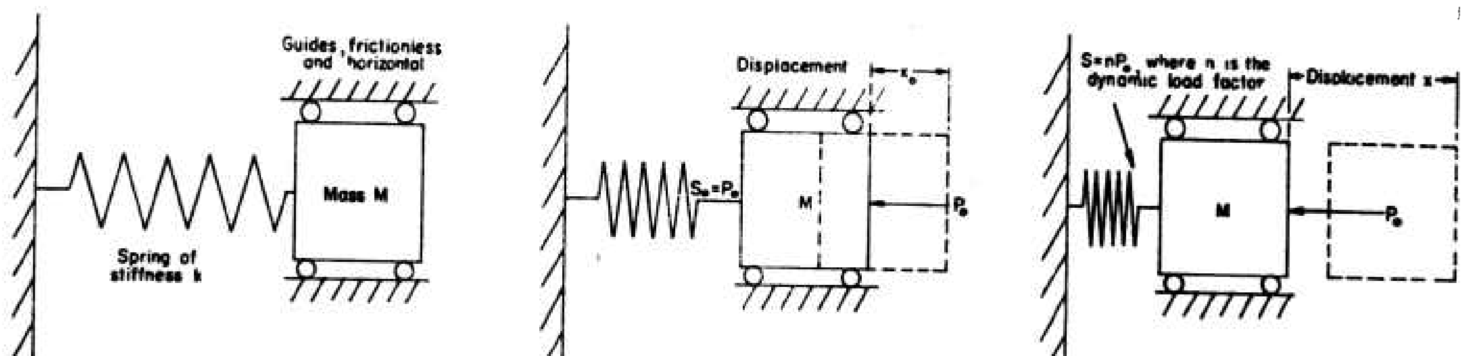
"All structures possess many degrees of freedom, but frequently one is so preponderant as to determine the behavior of the system, for all practical purposes. A structure is considered here which possesses a single degree of freedom; it is exemplified by a rigid mass, supported without friction or dam, attached to an inertialess spring, as shown in figure A (a).

"Figure A (b) illustrates the case of static loading on this system, in which the force P_o causes a displacement of the mass M and a shortening of the spring by the amount x_o. When the application of the force or load P_o is a function of time, as in figure A (c), dynamic loading occurs; the displacement x and the spring force S can then be conveniently related to the corresponding values for static loading by the use of certain nondimensional ratios.

"The ratio between the applied force P at any moment during the dynamic loading period and the maximum value of the force P_o applied to the system is designated as the disturbance factor, or more briefly, the disturbance. As P never exceeds P_o, the maximum value of this factor is 1.

"The static displacement x_o of the mass M under the steady load P_o can, as shown in figure A (b), be used correspondingly as a unit of displacement. Under sudden application of the force, P, the displacement rises to a dynamic value x, as in figure A (c); the ratio of this dynamic displacement x to the static displacement x_o is called the response factor, or, more briefly, the response. As shown by a comparsion of figures A (b) and A (c), the maximum value of this factor may greatly exceed 1.

"As the spring reaction S is assumed to follow Hooke's law, the response factor may be used to represent ratios of spring force, or load, as well as of displacement or deformation. Thus, the reactive force S exerted by the spring at any time is the maximum force P_o multiplied by the response factor at that instant. The numerical maximum of the response factor, derived from the ratios x/x_o or S/S_o, is the



(a) System having one degree of freedom

b) System showing static displacement x_o of the mass by a steady force P_o

(c) System showing dynamic displacement x of the mass by a disturbing force P whose peak value is P_o

Figure A. Undamped elastic system

dynamic load factor; it is the factor which, multiplied into the maximum load P_o, gives the maximum spring reactive force under the dynamic condition defined by the disturbance. Whereas S_o always equals P_o in static loading, the reactive force S exerted on and by the spring, or equivalent supporting structure, may greatly exceed the instantaneous value of the applied load P under dynamic loading conditions.

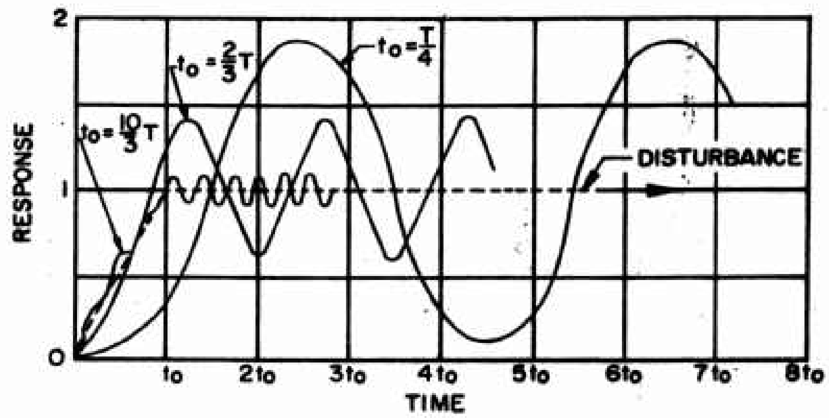
"The factor 2, to be found in all text books on mechanics for determining the force equivalent of a suddenly applied load, is a dynamic load factor. Knowledge of the dynamic load factor is a prerequisite to the design of a structure to resist a particular shock load.

"If the maximum load on the spring is such that the elastic range of the material is not exceeded, the stress in the spring will be at all times proportional to the reactive force S_o. Since it has previously been shown that the ratio S/S_o is equal to the ratio x/x_o, the response factor, expressed by the latter ratio, can also be used as a ratio of stresses in the spring material. Since the spring shown in figure A can be replaced by an equivalent elastic structure having one degree of freedom, such as a long, slender beam, it can be said that the maximum value of the response factor, which is equal to the dynamic load factor, gives the ratio of maximum stress in the beam under the shock or impact load to the stress set up in the beam by a static load P_o equal to the peak load P in the shock pulse, i.e., under the shock loading conditions.

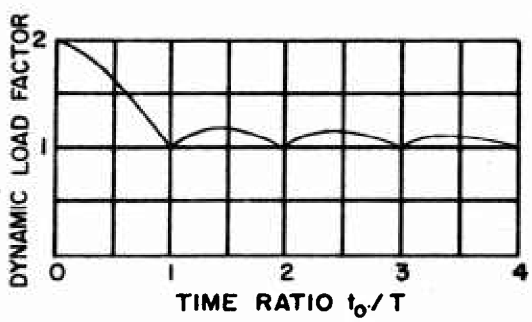
"Calculation of the response factor for several types of disturbance shows that the rate of application of the load and the duration of the load are of primary importance in a determination of the stresses set up in any particular structure. If the load is instantaneously applied, and remains constant for a duration exceeding half the natural period T of vibration of the structure, the dynamic factor is 2; that is, stresses are double those obtained on applying this load longer than is required to give the mass M in figure A a displacement of x_o, or to push it to a greater displacement x, and have it return to its initial position as shown in figure A (a), as it will do under the influence of a maximum reactive force S much greater than P. If the duration t_1, falls below T/2, then in all cases the dynamic load factor decreases from 2 to 0 as the ratio t_1/T drops to zero.

"If the load is not instantaneously applied, and if the time of rise t_o to peak load is less than one fourth the natural period T of the structure (figure B), then very nearly the maximum effect due to rate of application of load is realized; this is because the mass M in figure A is travelling to the left for more than the entire duration of the increasing push exerted by the load. As t_o becomes larger than T/4, the dynamic load factor progressively decreases.

"For a disturbance similar to the first half-cycle of the sine wave (figure C) , the largest dynamic load factor is for a duration t_1 of about one natural period, where the increase in stresses over the corresponding case of static loading is about 75 percent."

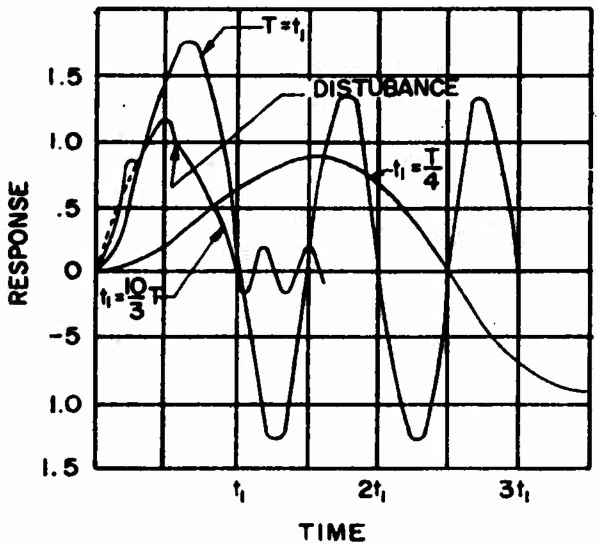


(a) RESPONSE OF VARIOUS SYSTEMS TO GRADUALLY APPLIED LOADS

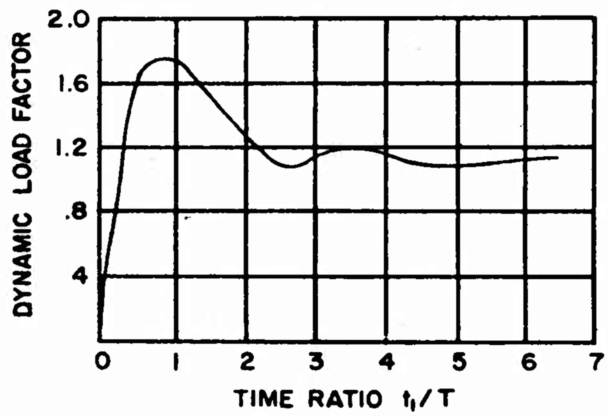


(b) DYNAMIC LOAD FACTOR FOR THE DISTURBANCE OF THE TYPE SHOWN IN FIG. a

Figure B. Disturbance applied gradually and maintained indefinitely



(a) RESPONSE TO SINUSOIDAL PULSE



(b) DYNAMIC LOAD FACTOR FOR A SINUSOIDAL PULSE

Figure C. Sinusoidal pulse

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